POWER

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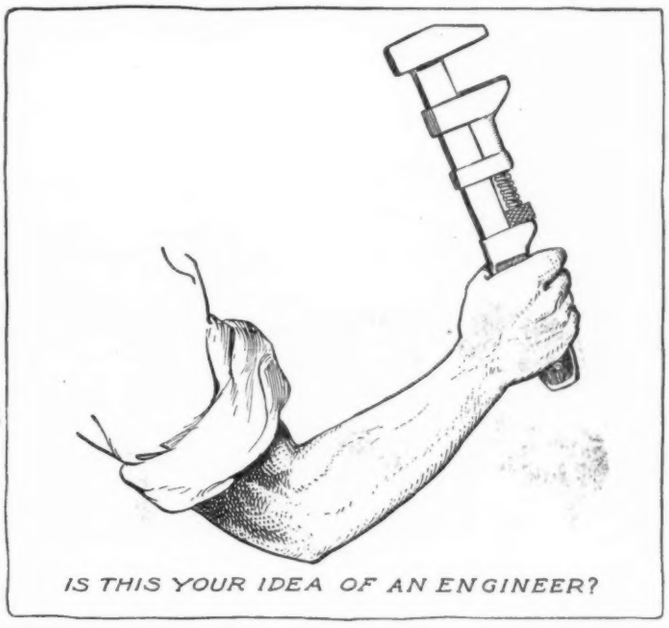
Hank Armstrong, Engineer (?)

By R. T. Strohm

They built a sawmill, some years back,
Acrost the way from Hackensack;
An' when they sought an engineer,
Hank Armstrong thought he'd volunteer.
Sez he, "I've ran a
mow'n machine
An' grindstone sence
I was thirteen,
An' sulky plow an'
corn drill, too."
They said they kinder
guessed he'd do.

He filled that b'iler
full's a tick
With water hauled
from Slocum's Creek;
An' then he took that
firebox
An' chucked it full of
hemlock blocks.
He throwed in coal oil
by the cup,
An' lit a match to start 'er up.
The flame shot out as he looked in
An' singed the whiskers off his chin.

That thingumbob that stands on top
Jumped open with an orful pop,
An' steam come shootin' in a cloud
Till Hank he said he kinder 'lowed
The thing was makin' such a noise
That it might rile the other boys.
An' so he twisted down the nut
Ontil the pesky thing stayed shut.



IS THIS YOUR IDEA OF AN ENGINEER?

We never thought Hank knowed so much
'Bout b'ilers, injines, belts an' such.
Why, if a bearin' box got hot,
He'd just splash water on the spot
Ontil the place got
cool, an' then
He'd start the whole
shebang again.
Or if a belt got slack
an' slipped,
He'd smear on tar
ontil she gripped.

Hank said he never
worked by rule,
'Cause any sort of
goldarned fool
Could use horse sense
an' mend a piece
With monkey-wrench
an' elbow grease.
Onct when the guvner
shaft got bent,
He grabbed his monkey-wrench an' went
An' hit the dingus one good smack
Acrost the side an' bent it back.

An' regular, twelve times a week,
The b'iler jints 'ud spring a leak.
Then Hank 'ud jump up off his bench
An' grab that doggone monkey-wrench,
An' go an' hammer every seam
Till it would quit its spittin' steam.
An' when we ast him, "Won't it bust?"
Hank grinned an' said, "In God we trust."

Deerfield River Hydro-Electric Development

By Warren O. Rogers

SYNOPSIS—A five-million-dollar hydro-electric development of New England. The generating stations, storage reservoirs, dams and tunnels are described.

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When Jonathan Catlin and James Ryder settled in what is now Shelburne Falls, Mass., some time between 1752 and 1756, and while treacherous Indians were roaming the forests and paddling their canoes upon the waters of what is now known as the Deerfield River, hydro-electric developments were unknown and undreamed of. The first water power of the town is said to have been developed by three men, David Fisk, Squire Barnard and an unknown, who operated a small grist and sawmill. This mill was doubtless located below the natural fall for which the town is named. The falls are of natural rock formation, as indicated by Fig. 1, which is a view of them as they appear at present.

Although the waters of the Deerfield River have turned the wheels of many industries, the development was along similar lines until a small hydro-electric station and dam was put in by the Greenfield Electric Light & Power Co. in 1904, at a point about one mile south of Shelburne Falls. The dam at this development is of modern concrete construction and the water is conveyed to the power house through a canal. The plant has a capacity of 1300 hp., and is operated in connection with the steam plant of the company at Greenfield, Mass.

three near Shelburne Falls, and the other about three miles north of Hoosac Tunnel, Mass., the railroad station on the Boston & Maine R.R. near the east entrance of the tunnel.

If one were to view the Deerfield River in July or August, the conclusion would be reached that it was a most undesirable stream for hydro-power-plant development, as the water supply would appear altogether insufficient to make such a proposition a paying investment. This river is the principal tributary of the Connecticut River and rises in the southwestern part of Vermont. If a map of Vermont and Massachusetts were examined it would be seen that the river has no particular source, but is formed by a number of small streams which drain a large area of southern Vermont, about 250 square miles. The only outlet of the river is through a narrow valley which runs south. The Deerfield River proper rises in the town of Stratton, Vt., and flows south, increasing in volume at East Branch Brook at Searsburg. This brook rises in the town of Somerset, Vt., and near its head waters at Peck's mill the first reservoir is being built. At Wilmington, Vt., the river is again augmented by the waters of the North Branch, and thence winds its way southwest until it crosses the Massachusetts boundary line; it then flows south through a narrow valley.

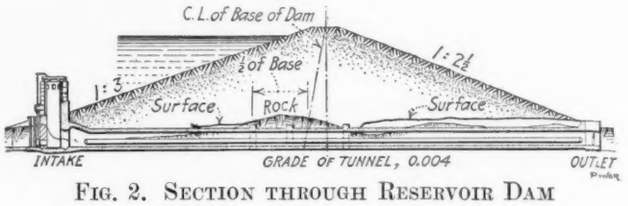


Fig. 2. Section through Reservoir Dam

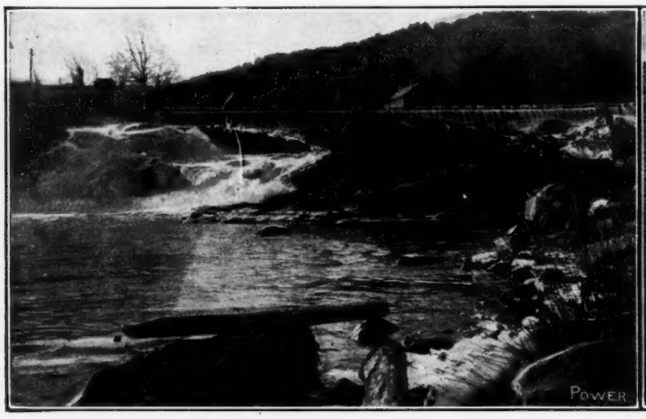


Fig. 1. New Dam at Shelburne Falls, Mass.

Fig. 3. Power House, Dam and Sluice Gates, No. 2 Development

However, the most interesting development from economic, hydraulic and electrical standpoints on the Deerfield River is that which has practically reached completion at or near Shelburne Falls. The company responsible for this work is the New England Power Co. The general contract for all of the work on the river is held by the Power Construction Co., whose headquarters are at Shelburne Falls. This company has done the engineering work for all the work on the river and has built with its own forces the storage dam and reservoir in the upper valley and all but one of the transmission lines. The four power plants have been built by subcontractors under the supervision and direction of the Power Construction Co.

At present there are four developments on the river;

The hills on either side have an elevation of from 800 to 2000 ft. above the sea level. In this narrow valley at a point midway between Munroe Bridge and Hoosac Tunnel, No. 5 development is being constructed. At the east side of the Hoosac Mountain the river runs southeast through a mountainous valley until it nears Charlemont, where the valley broadens out and the turbulent water

thence flows placidly to within two miles of Shelburne Falls, where the river again flows through a narrow valley. At this point the river makes a sharp turn around a hill about 600 ft. above the sea level and flows north through a deep ravine and then bends to the east and again to the south, receiving at this latter bend the water of the North River, which also rises in Vermont and drains an area of about 100 square miles, making a total watershed of approximately 520 square miles. It is at this horseshoe bend in the river that No. 4 development has been built. From the mouth of the North River the flow is almost due south through Shelburne

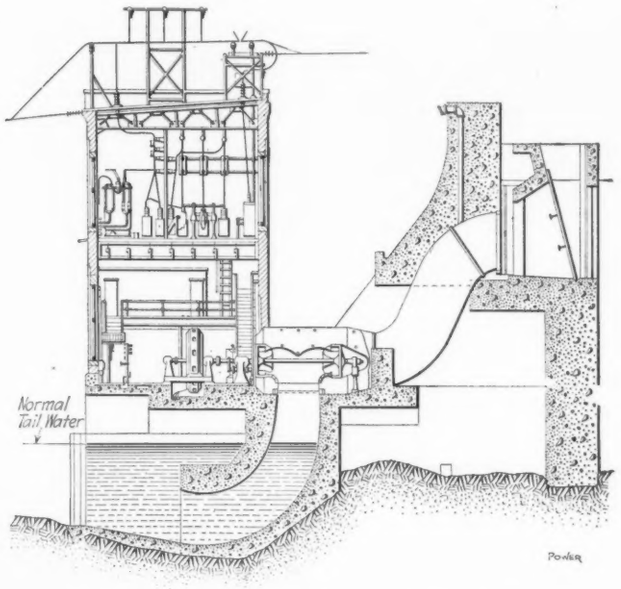


FIG. 4. CROSS-SECTION THROUGH NO. 2 PLANT

Falls, where the river makes an abrupt plunge over the falls to which reference has already been made.

From the falls the river flows southeast, the valley continues to be steep for about 10 miles, through which section a large number of power plants may be built as closely as the back water level of each will permit. It is in this section that Nos. 2 and 3 plants have been built and where No. 1, not yet begun, will be situated.

Although the flow of water during the dry summer months would not be sufficient to provide a maximum supply for the power plants, the deficiency is taken care of by the Somerset reservoir.

SOMERSET RESERVOIR

During the winter and spring the Deerfield River is frequently changed from the inconsequential appearing stream of the summer to a roaring, raging torrent. This

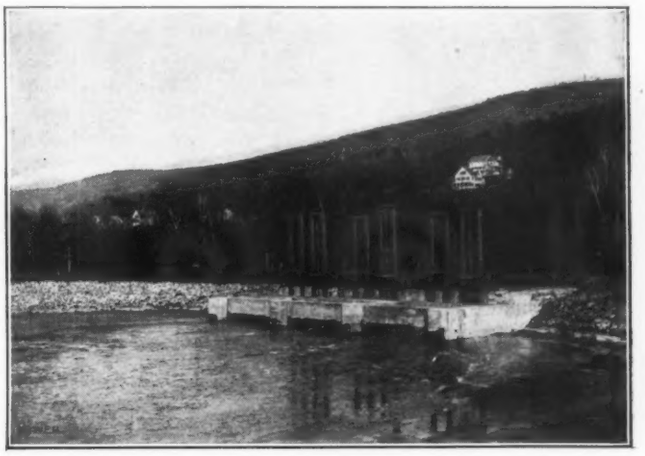


FIG. 5. FOREBAY OF NO. 3 POWER PLANT

water would flow over the dams without doing work if no provision were made to store it for use during the time that the supply of water from the drainage area was not sufficient to carry the maximum load of the power plants.

Therefore, the Power Construction Co. is building an immense storage reservoir at Somerset, Vt., near the source of the water supply, in which to store the surplus water in the winter and spring to be released during the time of low water. It will be seen how necessary this provision is, when it is understood that the monthly average discharge of water at Shelburne Falls ranges from 380 sec.-ft. to 2000 sec.-ft. in an average year and maximum and minimum flows have possibly reached 50,000 and 20 sec.-ft. respectively. Water from the east branch

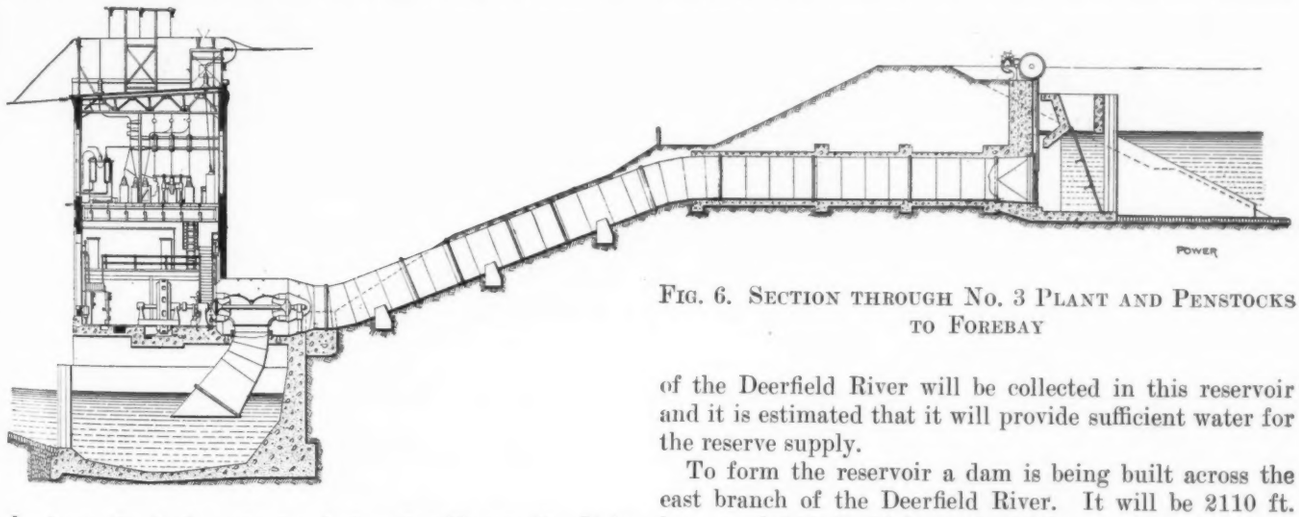


FIG. 6. SECTION THROUGH NO. 3 PLANT AND PENSTOCKS TO FOREBAY

of the Deerfield River will be collected in this reservoir and it is estimated that it will provide sufficient water for the reserve supply.

To form the reservoir a dam is being built across the east branch of the Deerfield River. It will be 2110 ft. long, and 110 ft. high at the highest point. The base will be 600 ft. wide at the maximum section and the flat top 20 ft. wide. The down-stream side of the dam will have a slope of 2.5 to 1, the up-stream being 3 to 1. The dam will be riprapped on both slopes. The reservoir

will have an area of about three square miles and will have an available storage capacity of approximately 2,500,000,000 cu.ft. with the water at the level of the spillway. The use of low flashboards on the spillway will increase the capacity of the reservoir by about 300,000,000 cu.ft.

The sluiceway and gates controlling the amount of water to be let down the stream during the dry season, will be near the east end of the dam. The discharge tunnel is 521 ft. long, in a trench cut in solid rock, concrete-lined and made in the shape of an inverted horseshoe, 11½ ft. high and 12 ft. wide. It will carry two 4-ft. delivery pipes through which the water will be discharged from the reservoir. In Fig. 2 is shown a cross-section of the earth dam.

A second storage reservoir, of about 4,500,000,000 cu.ft. capacity, will be built near the Massachusetts boundary line in Vermont and will be operated in series with the Somerset reservoir.

The Somerset dam will have a spillway about 200 ft. long, cut in the rock, and will provide an outlet for the reservoir in times of high water. It will be some time before this system of reservoirs will be in operation. Although the reservoirs and source of water supply are north of the power stations, the developments are numbered up stream. Thus, No. 1 power house and dam has not been commenced, while a few miles up the river from its probable site, No. 2 plant and dam is practically completed. A view of this work is shown in Fig. 3. Inasmuch as all of the generating stations are designed practically along the same lines, a later description of No. 4 development will answer for all three developments now completed. No. 5 plant is not completed and will contain differently designed generators.

No. 2 Development

This installation is about three miles below Shelburne Falls. The power house is built below the west end of the dam and very close to it. Water will be delivered to the three waterwheels through three 11-ft. steel penstocks. The dam has a maximum height of 90 ft., is 450 ft. long and has a spillway 276 ft. long. The base of the dam is 75 ft. thick and the top 10 ft. 6 in. wide. The spillway is of an ogee shape and extends from the headworks back of the power house, which headworks are 16 ft. higher than the dam proper to the east side of the river.

Short penstocks, connecting from the up-stream side of the dam with the waterwheels, have screens and gates. The dam has sluice gates, shown in Fig. 3, which facilitate the discharge of water. As shown, this station is at the bottom of a gorge formed by high hills on either side. The available normal head is about 58 ft. Fig. 4 is a cross-section through the plant and dam.

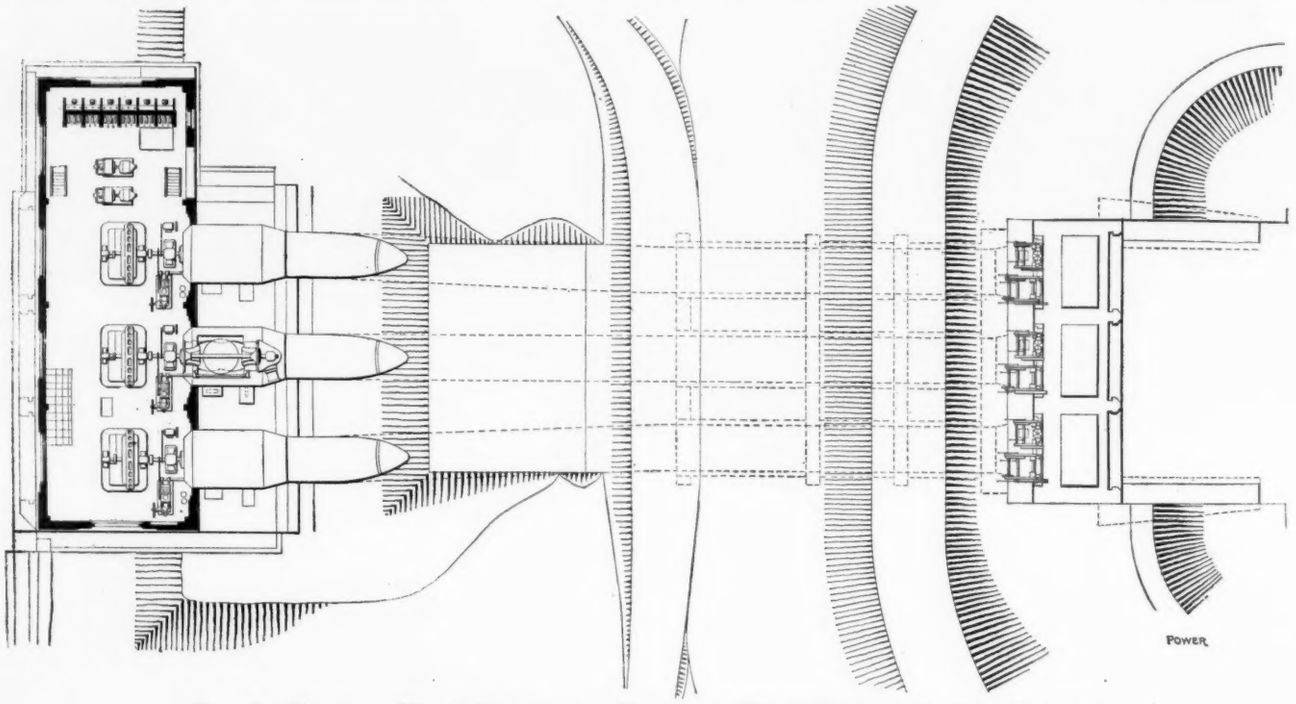


Fig. 7. Plan of No. 3 Plant and Penstock Connections to the Forebay

No. 3 Development

Station No. 3 is a short distance below the dam shown in Fig. 1. It is supplied with water from the storage supply above the dam by a conduit, canal and penstock. The new concrete dam has been built just downstream from the old timber crib dam, and is now 475 ft. long and 15 ft. high in maximum section. The intake delivers water to a reinforced-concrete conduit 12 ft. 6 in. by 17 ft. and 600 ft. long, which runs under a manufacturing establishment at the west end of the dam. The discharge is into a canal 920 ft. long, which terminates in a forebay, a part of which is shown in Fig. 5. The gates and concrete headworks are also shown. The gates control the water supplies to the three waterwheels through three 10-ft. penstocks, as shown in Fig. 6, which also shows a section through the plant. The Penstocks are each 160 ft. long and deliver water to the waterwheels at a normal head of about 64 ft. A plan view of the entire plant, penstocks and part of the forebay is shown in Fig. 7.

A view of No. 3 station from the top of the forebay is given in Fig. 8. Unlike Nos. 2 and 4, it is but one

story high for a part of its length and two stories for the remainder. The second story contains the oil switches, storage batteries and transformers. It will be noticed from Figs. 4, 6 and 8 that the casings of the water-wheels are placed outside of the buidlings; the same conditions obtain at plants Nos. 2 and 4.

No. 4 Plant

This installation is about 1¼ miles above Shelburne Falls. The power house is on the west side of the river and at the foot of a high hill. Owing to the topographical conditions at this point the dam was built on the other side of the hill around which the river winds. The dam is 240 ft. long at the crest and is built at an angle with the river to obtain a greater spillway to take care of high water in times of freshets. It is anchored on solid rock bed, and is 44 ft. high in maximum section and 10 ft. 6 in. wide at the top. At one end of the dam are three surface sluices, each 10 ft. wide, while at the other end is a submerged sluice 6x8 ft. controlled by a heavy cast-iron gate.

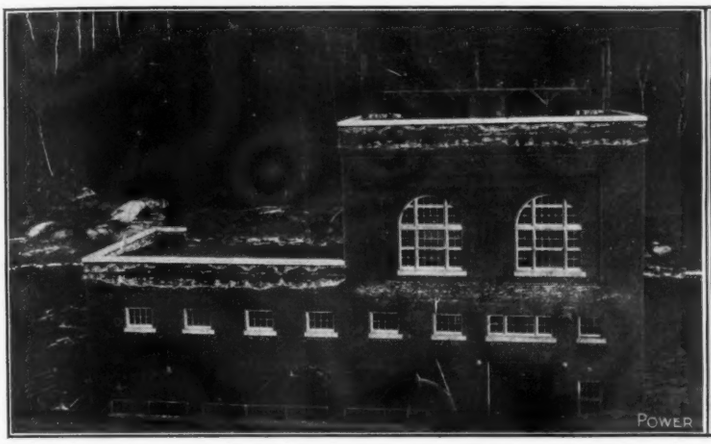


Fig. 8. Penstocks of No. 3 Power Plant

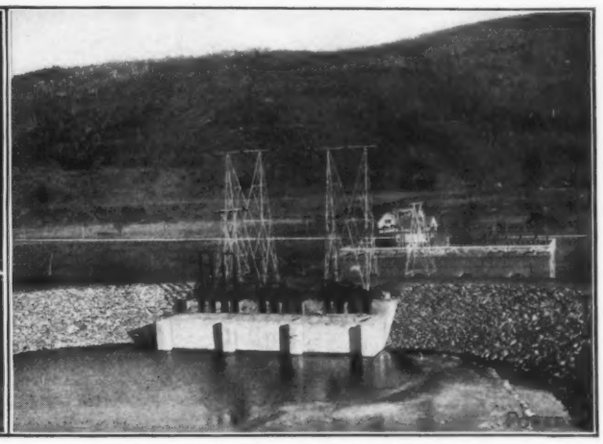


Fig. 9. Forebay of No. 4 Development

Although the river flows north, east and south in getting past the hill under which the power house is located, the tunnel from the dam intake to the forebay is made in a straight line and is 1518 ft. long, the power house being directly opposite the first bend in the river, but on the opposite side of the hill.

Like those at the other stations, each penstock has two gates, each 7x12 ft. in size. Fig. 9 gives a view of the forebay and head gates. The power house is shown beyond, and below the level of the forebay. This building differs from the No. 3 building, in that it is two stories high throughout, and the switches and transformers occupy the entire upper floor.

This station is perhaps the most interesting on account of the tunnel work. The concrete-lined tunnel is in the form of a horseshoe and has an area equal to a circle 13 ft. in diameter. It is built with its highest point in the middle to facilitate drainage during construction. The tunnel extends from just above the dam to the forebay on the east side of the hill, the tunnel intake is controlled by a steel gate weighing about 34,000 lb., operated by a 15-hp. motor. The three penstocks are each 10 ft. in diameter and of riveted steel, varying in thickness from ¼ in. at the top ends to $\frac{5}{16}$ in. at the bottom. The turbines discharge through draft tubes into a tail-race which discharges directly into the river, as shown in Figs. 4 and 6.

Work on the tunnel was started in December, 1911, and five months later the hole through the hill was made. The amount of rock excavated is estimated to have been 12,000 cu.ft. With the exception of about 120 ft., the bore is through solid granite. In Figs. 10 and 11 is shown the concrete work at the inlet and outlet of the tunnel. Some idea of the amount of work necessary to make the forebay can be gathered from Fig. 12. The forebay is about 200x225 ft. at the top and 50x75 ft. at the bottom, and is 33 ft. deep. The basin is of earth with riprap lining, and will impound approximately 2,000,000 gal. of water.

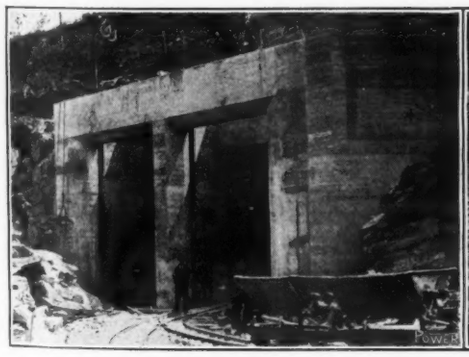


Fig. 10. Tunnel Intake Practically Completed

Fig. 11. Tunnel Outlet Concrete Work

Fig. 12. Tunnel Outlet for Forebay Under Construction

Equipment of Power Plants

As already stated the equipment of the three stations is practically identical, both hydraulically and electrically. Each of the three stations at Shelburne Falls contains three 2000-kv.-a., 2300-volt, three-phase, 60-cycle General Electric generators, each directly connected by a horizontal shaft to a 3200-hp. double runner, central discharge Wellman-Seaver-Morgan turbine, running at 257 r.p.m. The speed of each unit is governed by a Lombard special horizontal governor, directly connected to the turbine-gate shaft, and belt-driven from the generator shaft. All governors have a 30,000-ft.-lb. rating. The governors operate under oil pressure supplied by a 4x6-in. triplex, plunger pump. The generators are installed along one side of the generator room, as shown in Fig. 13. The governors are placed between the generators and the inside end of the turbine casing, which extends slightly into the generating room. Fig. 14 shows this feature.

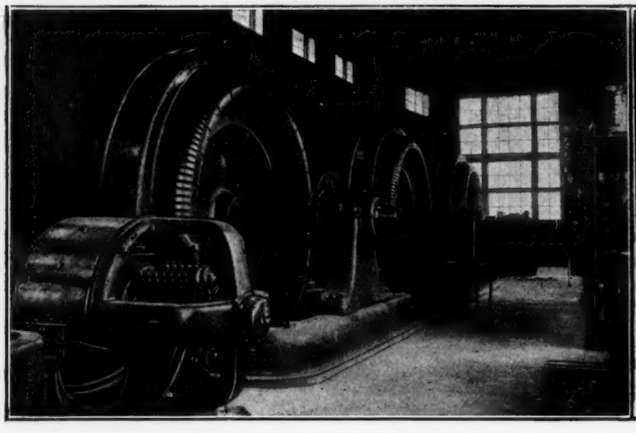


Fig. 13. Generator Room

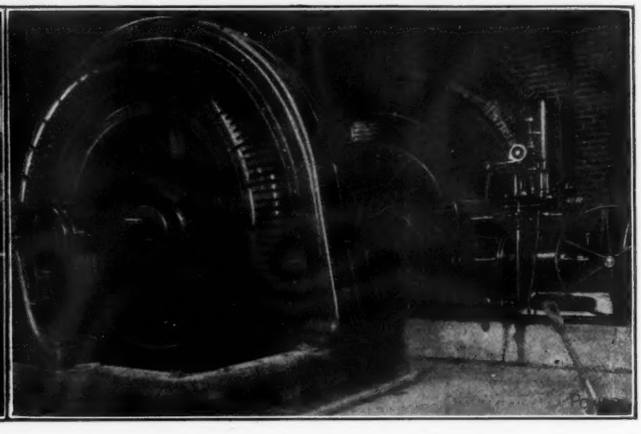


Fig. 14. Special Design of Governor

Each plant has two exciter units, but none is driven by an individual waterwheel. Each unit consists of a 150-hp., 2300-volt induction motor driving a General Electric 100-kw., 125-volt generator. The reason for depending upon motor-driven exciters is because each station is tied to the others and also to the Vernon, Vt., power plant, described in the Sept. 28, 1909, issue of Power. There will doubtless never be a time when electrical energy will not be available to operate the induction motors.

Transformers and Oil Switches

There are three 300-kv.-amp. water-cooled, oil-insulated transformers in each station, transforming from 2300 to 69,000 volts, placed on the second floor of the station. The oil switches, of the remote control type, have a rating of 150 amp. per terminal at 70,000 volts, and are connected overhead through the 69,000-volt buses to the transformers, occupying the second floor.

All the transformers rest upon rails leading to an opening in the floor at one side and midway of the building, which permit of lowering a transformer to the floor below by a 30-ton chain-block hoist, where it can be handled by a hand-operated traveling crane.

Cooling water for the transformers is supplied by direct connection to the penstocks. In an emergency the oil in the transformers can be discharged into the river through a 4-in. pipe, by operating a quick-opening valve. Fig. 15 shows a view of the transformers and oil switches.

On the opposite side of the generating room, midway of the building is a 10-panel operating switchboard. It controls the three generators, two exciter units, two 3000-kv.-a. transformers and the outgoing 66,000-volt lines of which there are two from No. 3 station and four from Nos. 2 and 4. Each station has a storage-battery system for controlling the oil switches.

Transmission Lines

In Figs. 6, 8 and 9 may be seen how the high-tension wires are suspended on the station roof. All lines are dead-ended on strain insulators fastened to the station wall. From these taps are carried to insulators supported on a steel framework. The lines are spaced horizontally, about 8 ft. apart, and are carried 15 ft. above the roof.

All transmission lines now under construction comprise two three-phase circuits of Nos. 1 or 2/0 cable. One line runs from a junction point opposite No. 4 station to the Vernon station on the Connecticut River, a distance of 16 miles. Another double-circuit line of No. 1 copper cable extends 15 miles from the No. 4 station to No. 5 station, which will be described later. There is also a two-line double circuit extending from the junction point at No. 4 station down the river to No. 2 sta-

Fig. 15. Transformer and Oil Switches, No. 4 Plant

tion with a ½-mile tap to No. 3 station. From station No. 2 a double-circuit line of No. 00 cable extends 60 miles to Milbury, Mass., and a tie line runs from the latter point to the Greendale substation, 14 miles away, and a short distance outside of Worcester, Mass. There are also the double three-phase lines of the Connecticut Transmission Co., extending from Vernon, Vt., to Worcester, Mass., with several branches. Fig. 16 is a map showing the existing circuits in dark lines and the proposed circuits in dotted lines.

No. 5 Station

Development No. 5, which will operate in connection with the same series of power plants, is not completed. The station is a brick building, like the other power stations, and is about three miles from the Hoosac Tunnel station of the Boston & Maine R.R. It is reached by the Wilmington & Hoosac R.R. or by traveling over a winding road which follows the Deerfield River. The initial equipment of the station will include three 3000-kv.-a. transformers which will reduce the potential from 66,000

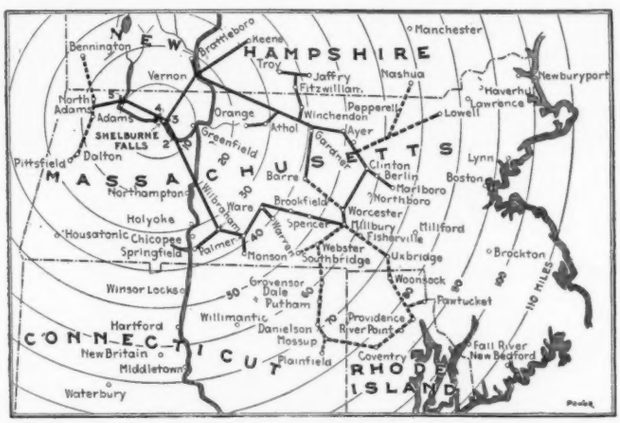


Fig. 16. Showing Existing and Proposed Electrical Circuits

volts to 2300, three 150-kw. exciter units, switching apparatus, lightning arresters and three frequency changers, each consisting of a 4250-kv.-a., three-phase, 60-cycle, 2300-volt generator directly connected to a 4000-kv.-a., three-phase, 25-cycle, 11,000-volt alternator or approximately 3000 kv.-a., single-phase. These 60-cycle generators will operate as synchronous motors to drive the 1100-volt generators which will supply power to operate the electric locomotives through the Hoosac Tunnel. These are at present operated from the Zylonite steam-power plant described in Power, July 4, 1911. When the waterwheel-driven generators are installed, of which there will be three, these two machines can be run as generators. This arrangement will also give a flexible interconnection between the systems and will permit of the Zylonite plant feeding back into the hydro-electric system in case of water shortage.

The waterwheels to be installed will be of 6000 hp. capacity and will be directly connected to the frequency changer sets above described, generating either single- or three-phase energy. The machines will be able to operate in parallel with those of the Zylonite plant in either single- or three-phase service. To meet the requirements of the heavy freight traffic through the Hoosac Tunnel the 25-cycle machine in each unit will be capable of developing 4300 kv.-a. for 20 min. of each hour at 70 per cent. power factor. During the remaining 40 min. of each hour the machine will be capable of operating at a load of 2000 kw.-a. at 90 per cent. power factor.

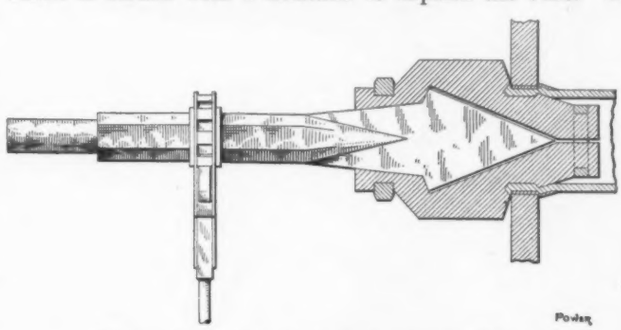
According to plans as now formulated the water to the turbines of No. 5 plant will come to them with a head of 200 ft. The dam will be about 2½ miles above the present station and the water will be conveyed to the top of the hill back of the power house through an alternate canal and concrete conduit, emptying into a forebay from which the water will come to the waterwheels through penstocks. Plans have also been formulated to some extent for another and larger plant, when the demand for electrical energy occasions its construction.

No. 5 power house is of brick and is 100x100 ft. and 60 ft. high. Large windows afford the admission of sufficient light. The concrete foundation extends 20 ft. below the water line and contains 3500 cu.yd. of concrete.

The developments above described will cost $5,000,000. The waters of this very erratic river have been made to produce electrical energy to be transmitted to distant cities and towns. In addition to the power generated by the stations of the New England Power Co., on the Deerfield River, there is a plan for additional water power controlled by the Connecticut River Co., to tie on to this system and form a combination capable of generating an ultimate capacity of 200,000 hp.

Moore Tube Expanding Beading Tool

Illustrated herewith is a section of a flue sheet and a flue and shim expander. The expander comprises two working members, held together by elastic bands at each end, forming an elliptical cylinder. A shoulder is formed for the enlargement inside the flue sheet. Lugs are provided to form the beading curve and between the lugs and the shoulder is the expander. The inner surfaces of the working members conform to the shape of the wedge which is struck with a hammer to expand the tube. A



Tube Expanding and Beading Tool

ratchet is provided for slowly rotating the device. The operations of placing, rolling and beading the tube are performed at the same time by the one tool.

It is handled by F. C. Worth, 310 Monadnock Block, Chicago, Ill.

Two cubic feet of water falling 6 ft. will produce 1 hp. of energy in the most modern electrical machinery. The total discharge of the Mississippi River into the Gulf of Mexico annually is 18,400,000,000,000 cu.ft.

Electrical Energy Meter

This device is designed to measure the energy of steam and gas engines. With it all the positive forces are added, the negative forces are subtracted therefrom and the net active forces measured. This is accomplished by applying the excess pressure of one side over that of the other side of the piston of the engine, to determine the voltage and direction of a current of electricity generated in a magneto by the varying motion of the piston in proportion to the pressure and motion, which determines the energy of the engine, and thus measures the energy developed by the engine.

In Fig. 1 is shown a longitudinal section of an engine cylinder and a top view of the magneto, also a front view of the meter forming a part of this device and details of other parts later referred to. Fig. 2 shows the device attached to a steam-engine cylinder.

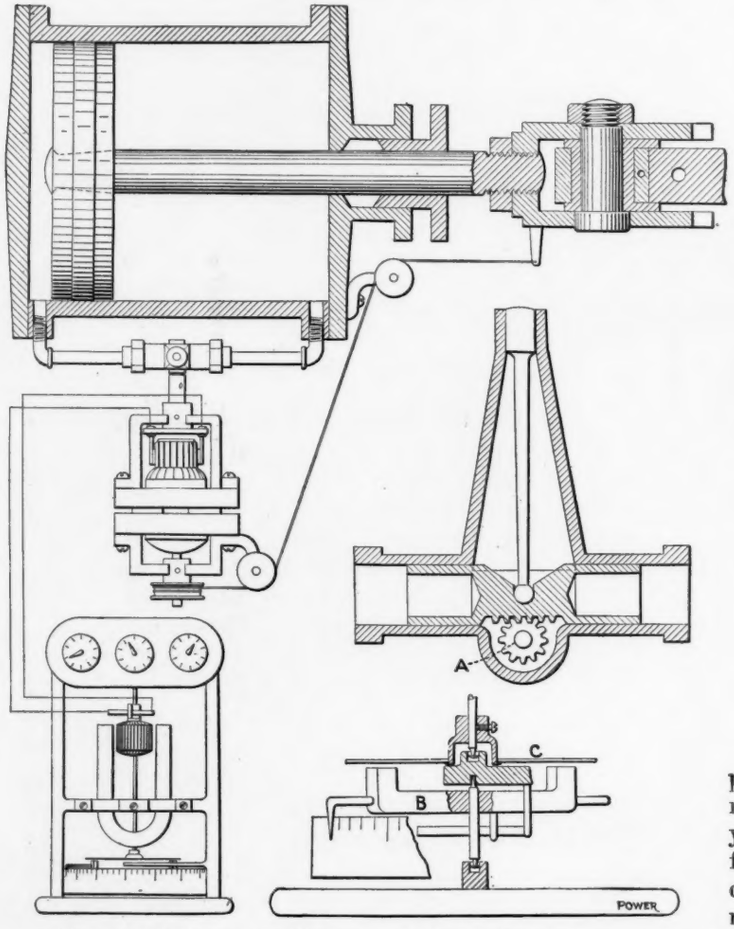


FIG. 1. DETAILS OF METER AND CONNECTIONS

Referring to Fig. 1, a stud is rigidly fixed in the crosshead to which a cord is attached leading to a pulley on the shaft of the magneto.

To one side of the engine cylinder an auxiliary cylinder is mounted, an enlarged vertical section of which is shown in Fig. 1. The ends of the piston in this cylinder are put in communication with the corresponding ends of the engine cylinder by two tubes. The auxiliary cylinder has a casing forming a chamber, in the top of which a spring is rigidly secured with its lower end engaging in a socket in the piston. The chamber in the base of the cylinder contains a shaft carrying a pinion *A* that engages with a rack integral with the piston and operated by it.

The shaft projects from the end of the chamber and into a hollow shaft which forms a part of the magneto. Inside of the magneto pulley the outer end of a spiral spring is fastened; the other end is secured to one end of the bearing. The cord, by the movement of the engine piston, rotates the magneto pulley and armature in one direction, and the spring rotates the armature in the opposite direction to correspond with the reciprocatory motion of the engine piston and crosshead. While the armature is rotating in one direction, the pressure of the steam against the auxiliary piston forces it forward against the action of the spring an amount in proportion to the varying pressure in the cylinder, thus rotating the pinion *A* in, say a clock-wise direction. This motion is communicated to a movable brush-yoke on the magnet shifting the brushes away from the neutral point on the commutator in a clock-wise direction. As the voltage of the magneto at any given speed of the armature is proportional to the displacement of the brushes from the neutral points, and as the voltage of a magneto, for any given location of the brushes, is in proportion to the speed of its armature, the voltage at any instant will be in proportion to the combined effect of the location of the brushes with respect to the neutral points and the speed of the armature; these two conditions are governed by the varying steam pressure in the cylinder and the varying speed of the engine piston at the same instant. Therefore, the voltage will be in direct proportion to the pressure and motion which determine the horsepower of the engine.

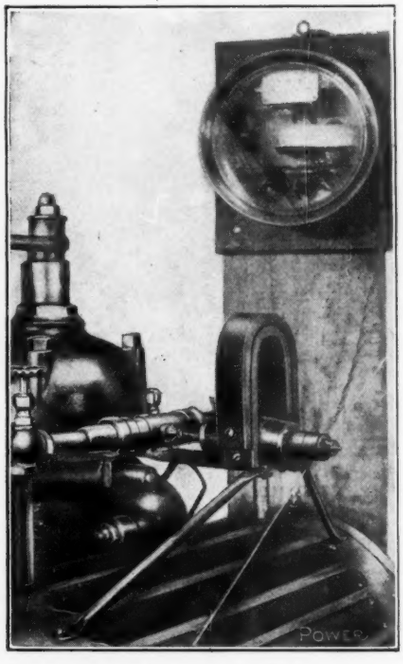


FIG. 2. METER ATTACHED TO AN ENGINE CYLINDER

Having completed this stroke, the engine piston starts in the opposite direction and the pressure, introduced on the opposite side of the piston, is exerted against the opposite side of the auxiliary piston, moving it in the op-

posite direction, which motion is communicated to the brushes. The direction of the armature rotation also having been changed, the current from the magneto will be in the same direction as before.

Upon this return stroke, after cutoff, when the piston reaches the compression point, the compression pressure enters the auxiliary cylinder at the opposite end and reverses the piston, which moves the brushes to the opposite side of the neutral point before the direction of rotation of the armature is changed, and the current from the magneto will be in the opposite direction, having the effect of reversing the meter, which is wired in circuit with the magneto. Compression is taken into account because it does not represent active work, but is negative work and tends to retard the engine speed at the end of each stroke.

The meter, by which the horsepower-hours are registered and the horsepower is indicated, consists of any ordinary form of decimal wheels (not shown) and a dial plate which registers the total horsepower-hours. The meter is electrically connected to the magneto. The magnetic field against which the current delivered to the meter armature from the magneto acts, causes that armature to rotate in the direction in which the register adds when the current is in one direction, and in the direction in which the register subtracts when the current is in the opposite direction. The magneto furnishes current in one direction, when the pressure in the auxiliary cylinder is doing work, and the current will be in the opposite direction when the back pressure or compression in the engine cylinder is greater than the forward pressure. Therefore, the net result of all these conditions is that the meter will register the work done by the engine.

To directly indicate the horsepower of the engine there is a dial in the bottom of the meter and a magnet *B* secured to a lower shaft, as shown in the lower right-hand detail in Fig. 1. An index is secured to one end of *B* and indicates the horsepower developed by the engine on the dial. One end of a hair spring is secured to the magnet *B;* the other end is secured to the bottom pivot block. This tends to hold the magnet and the index in a zero position on the dial.

As the upper meter shaft is rotated from left to right, the forward direction of the meter, the magnetic torque, or pull, between the magnet *B* and the disk *C*, which is secured to the shaft, tends to rotate the magnet *B* in the same direction against the action of the hair spring and the deflection will be in proportion to the speed of the disk. This is in proportion to the power developed, and causes the index to indicate upon the dial the horsepower of the engine at any time.

This meter registers only the net or working horsepower of the engine after having subtracted all of the negative, exhaust or counter forces. The meter may be set in the office of the superintendent where it can be consulted at any time.

This energy meter is manufactured by the Loetscher-Ryan Manufacturing Co., Dubuque, Iowa.

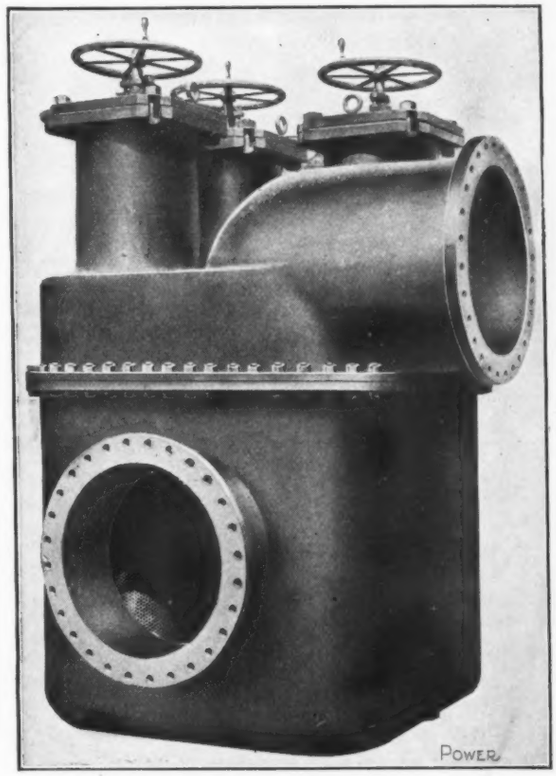
❖

The smallest dynamo in the world is said to be the one now on exhibition in France, writes a New England engineer contributor. It is smaller than an American penny in diameter and weighs only one-fifth of an ounce, is 6 in. high, and used as a generator as well as a motor. It consumes 2 amp. at a pressure of 25 volts. The curious part of the whole thing is that it is accurately made, and can be operated by a small pocket battery.

Large Lagonda Multiple Angle Type Water Strainer

One of two large multiple water strainers recently built by the Lagonda Manufacturing Co., Springfield, Ohio, is shown in the accompanying illustrations. The Southern California Edison Co. will use the strainers to remove fish, sticks, leaves, seaweed, ice, etc., from the water supply to its power plant.

The strainers are not only of large size (30 in.) but are of the angle type, the water passing out in a direction 90 deg. from the inlet. Each strainer contains three filtering baskets, each having a free straining area equal to that of the pipeline. When one of the baskets is being cleaned, the two remaining baskets have an area



MULTIPLE ANGLE-TYPE WATER STRAINER

at least twice that of the pipeline. Thus there is little resistance to the flow of water or loss of head caused by friction. The low velocity in the strainer also permits impurities to settle, and eliminates wear caused by the erosive action of water moving at high velocity.

As the entire strainer can be set beneath the power-plant floor, only the handwheels and top of the bonnets project above. Any one of the individual baskets can be removed from the body of the strainer by turning one of the handwheels.

❖

Extensions of the Niagara Falls Hydro-electric plants on the Canadian side are at present being pushed with the view of utilizing all the water available for power purposes under the existing treaty. The Toronto Power Co. will enlarge its output to 125,000 hp., and a new line, operating at 85,000 volts, is being built between Niagara Falls and Toronto. The plants on the American side of the Falls are at present using all the water allotted to them by the Secretary of War and the granting of additional water up to the amount at the disposal of the United States under the present treaty is optional with the Secretary.

Flow of Water over Weirs

BEN D. MOSES

Often in practice the weir is used to measure water and its use is purely mechanical to many of its advocates, as the quantity is found generally by looking up the values in tables, picking them off of curves, or substituting in formulas. Unless, then, one is more or less familiar with the theory of the weir itself, quite distressing mistakes are likely to be made.

Nothing is prettier, or will hold the attention as will a sheet of water falling over a well made weir, whether it is small, and used in the laboratory or one spanning a river valley, unless it is the theory of the flow of the water. Then, too, the actual results obtained in the field conform so closely to what the theory predicts that the possession of one begets a desire for the other.

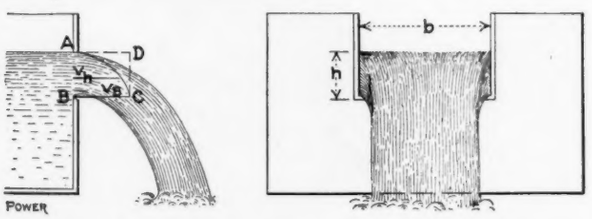


FIG. 1. RECTANGULAR WEIR, SHOWING VELOCITY CURVE

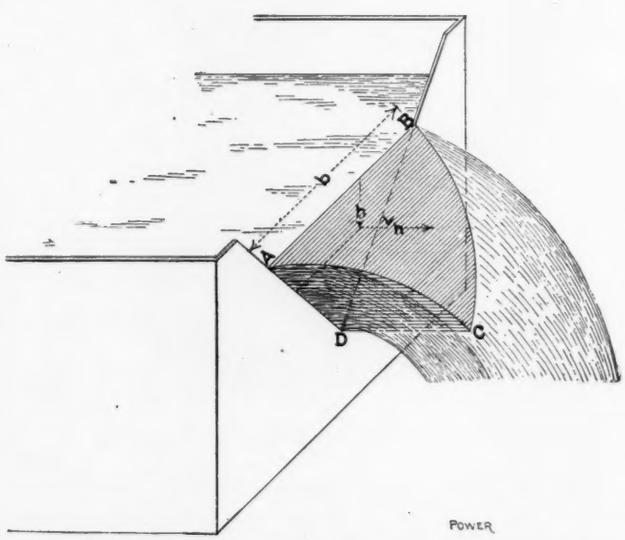


FIG. 2. TRIANGULAR WEIR, SHOWING VELOCITY CURVE AND VOLUME

In the use of the weir for water measurement, several things must be assumed as understood:

(1) The velocity of water flowing through a free orifice under a head h, theoretically equals $\sqrt{2gh}$, where g is the value of the earth's pull, or equals 32.2. In other words, if the pressure of a column of water h is converted entirely into velocity, the velocity would be equal to $\sqrt{2gh}$.

(2) Because of friction, eddying and irregularity of flow, the actual velocity will be something less than the theoretical; and,

(3) Because of contraction, the cross-sectional area of the stream will be something less than the area of the notch over which the water is flowing. This phenomenon will effect the quantity Q directly, so that in place of Q being av, it will be some constant c times av.

Keeping these things in mind, consider a stream flowing over, first a rectangular weir h deep and w wide.

RECTANGULAR WEIR

The velocity of a particle of water at A (Fig. 1) will be zero, and at any depth h below A will be equal to $\sqrt{2gh}$.. Now the abscissas of a velocity curve AC would show the velocity at any depth, while the quantity Q in one second is the volume of a solid having the cross-section ABC and length b or Q equal to the area $ABC \times b$. But the curve ABC is a parabola, since its abscissa at any point is numerically the square of the ordinate, and the area of a part of any parabola is $\frac{2}{3}$ of the area of the rectangle formed by the ordinates and abscissas of the points.

Therefore:

$$Q = \tfrac{2}{3} \text{ area } ABCD \times b$$
$$= \tfrac{2}{3}\, bh \times v = \tfrac{2}{3}\, bh \times \sqrt{2gh}$$
$$= 5.35\, bh^{\frac{3}{2}}$$

This is, however, the theoretical Q; the actual Q has been found to vary from 0.59 to 0.64 of this value. This constant, however, depends upon the condition of the weir and must be determined in any given case.

In general $Q = K\ 5.35\ bh^{\frac{3}{2}}$, where K is the experimental constant.

TRIANGULAR WEIR

A similar consideration of the velocity curve (Fig. 2), shown shaded, of the triangular notch brings us to the point that,

$$Q = \textit{Volume } ABCD$$

but the volume $ABCD$ can be shown to be

$$\text{Volume} = \tfrac{8}{15}\, K \sqrt{2gh}, \text{ where } K \text{ is the ratio } b/2h$$

Reducing this now, for the 90-deg. notch, which is the one used most in practice, where $K = 1$.

$$Q = \tfrac{8}{15} \sqrt{2gh^{\frac{3}{2}}}$$

and introducing the constant for contraction, which for the triangular notch is fairly constant and equals about 0.6

$$Q = 2.56\, h^{\frac{5}{2}}$$

In the above deductions it will be understood that the shaded part representing Q is not the form of the water falling over the weir, as the water is affected by the gravity of the earth. While the above curve is purely a velocity curve, it represents to scale what the ideal case would be if gravity ceased to act the instant that the water passed the knife edges of the weir.

It might be well here to say that often in practice suppressed weirs are used, in which case Francis' formula is generally applied.

$$Q = Cbh$$

in which C is dependent upon the number of contractions.

[For a more extensive treatment of the mathematics of the flow of water over weirs, through notches and apertures, the reader is referred to a series of articles on hydraulics in POWER for 1908-9.—EDITOR.]

ELECTRICAL DEPARTMENT

Motors and Motor Applications

By A. B. Morrison, Jr.

The use of electric motors for driving all classes of machinery is so common that little thought is often given to the conditions under which the machine is to operate or to the characteristics of the motor to be applied to it. A careful study of these two points often discloses the fact that a motor of an entirely different size or type from that first planned should be installed. It is the intention of the present article to show in a general way the important factors which must be considered in deciding the proper size and type of motor to install, then to discuss the operating characteristics of the various types, and finally to indicate the best type for certain classes of work.

To secure the most satisfactory results from motor drive it is essential that the type, as well as the size of motor, be properly adapted to the work contemplated. This is especially important in the case of individual drive where the shortcomings of the motor are more serious than in group drive. Even though the size of the motor may be ample to operate the machine under normal load the installation may be very unsatisfactory because the motor does not develop the proper starting torque or else requires excessive current at starting, the speed regulation may not be suited to the machine or some other inherent fault shows up which makes that particular type of motor unsuited for the work required. While, generally speaking, in many installations there cannot be a choice between direct and alternating current, yet the kind of current available is often not important provided the correct type of motor be selected.

In considering the application of a motor it is necessary to take into account the following, all of which must be carefully analyzed if satisfactory results are to be obtained:

Starting torque.
Maximum torque.
Cycle of operation.
Method of drive.
Type of machine and installation.

It is important first to understand clearly the difference between horsepower and torque. The former is the rate of doing work, while the latter is only one of the quantities making up horsepower. The torque of a motor is sometimes defined as the pull or force exerted at the surface of the armature, multiplied by the radius of the armature. For commercial purposes, however, it is defined as the pull exerted at a certain radius from the shaft center. For convenience this pull is usually expressed in pounds and the radius in feet, which, multiplied by the peripheral speed, gives an expression in foot-pounds which is readily reducible to horsepower. Assuming that the force is applied at a distance of one foot from the center of the shaft so that r (radius) $= 1$, then

$$Torque\ (T) = \frac{hp. \times 33{,}000}{2\pi \times r.p.m.} = \frac{hp. \times 5252}{r.p.m.} \qquad (1)$$

From this it is evident that for a given motor and a given horsepower, the torque varies inversely as the speed. If the first definition of torque is assumed, that is, the force acting at the surface of the armature—it is apparent that the torque would be dependent on the diameter of the armature as well as the speed; whereas, by the second definition, it is independent of the armature diameter. In motor applications one is concerned with the torque exerted on the motor shaft and not at the surface of the armature.

Starting Torque

With a machine having heavy inertia, the required starting torque may be in excess of that required to keep it running after once in operation. For example, a punch having a heavy flywheel or a long lineshaft carrying a number of large pulleys with heavy machines belted from them, is harder to start than to keep up to speed after once in operation. It is possible to approximate the starting torque by fastening a lever to the driving shaft of the machine and measuring the pull in pounds necessary to turn the shaft; this pull multiplied by the radius in feet, measured from the center of the shaft to its point of application gives the torque.

Assuming a concrete case, suppose it requires a torque of 50 ft.-lb. to start a certain machine; also to keep the machine running after it is once up to speed requires 7.5 hp. Conditions may be such that a motor with a full load speed of 1750 r.p.m. can be used. By formula (1)

$$Torque = \frac{7.5 \times 5252}{1750} = 22.5\ ft.\text{-}lb.$$

Hence, to start the machine, the motor must exert a starting torque equal to $50 \div 22.5$, or 222% of normal. An ordinary squirrel-cage induction motor, if the installation were alternating current, would probably not be suitable because of insufficient starting torque.

If a 1150-r.p.m. motor be considered in place of 1750 r.p.m., the torque becomes

$$\frac{7.5 \times 5252}{1150} = 34.2\ ft.\text{-}lb.$$

and the starting torque becomes $50 \div 34.2$, or 146% of normal, for which starting torque, the average squirrel-cage motor of this rating is satisfactory.

From the foregoing it is apparent that if a 1750-r.p.m. induction motor were required because of special conditions, it would be necessary to use a type which would give greater starting torque, either a motor having wound rotor or with a high resistance rotor. Furthermore, the time element is important as regards starting torque, for if the maximum starting torque is required for some time (the machine is slow to start), a type of motor which might answer the requirements if the time of starting were short, would not answer for the longer time.

Maximum Torque

With a heavy planer the torque required at the moment of reversal may be sufficient to stall or to slow down the

motor. Similarly with a heavy punch, in spite of the flywheel, the motor may develop insufficient torque to keep up the speed. The maximum torque is, in a sense, a measure of the overload capacity of the motor, particularly for short periods, although this is not strictly correct, as the maximum torque is developed at a lower speed than that at which the maximum horsepower exists. For longer overloads the heating is, of course, the limiting feature. In cases where a motor is large enough for starting and normal operation, but not large enough for the maximum overload required for perhaps only a second or two, the addition or substitution of a suitable flywheel will sometimes cut down the maximum torque required. In some cases it may be necessary to install a motor larger than necessary for the average work simply to secure the maximum torque necessary to keep the machine up to speed at all parts of the cycle.

Cycle of Operation

The cycle of operation required of a machine driven by a motor has an important bearing on the proper size to install. The allowable load is almost always limited by the permissible heating, which is proportional to the square of the current which the motor draws from the line, and since this current is a measure of the horsepower the heating is proportional to the square of the horsepower and to the time. In some cases, however, the maximum load may exist for only short periods and while a motor of proper size to drive the machine during the cycle would be large enough so far as heating is concerned, it might not be large enough to handle the maximum load nor to develop the starting torque necessary.

Method of Drive

A motor may be connected to a machine directly through a coupling, by a belt or chain, or by gears. Where a clutch coupling is used it is possible to bring the motor up to speed and then throw in the clutch. This is advantageous where the driven machine may require an excessive starting torque. A direct-connected, geared, or chain-driven motor is subject to all the fluctuations of load of the driven machine, whereas with a belt connection or clutch there is a chance for slippage. This question must also be considered in the case of a number of machines which may be group driven from one shaft or may be individually driven. The choice of group or individual drive may depend largely on the starting and maximum torque requirements of the various machines. If these are excessive they may be better handled by one large motor which has a large starting torque compared to that required by each separate machine, starting the machines one after the other, rather than by a number of smaller motors, built specially for the work.

In the case of alternating-current motors, especially, the question of direct connection or some other form of drive is important as the characteristics of the induction motor are modified considerably by the speed.

Type of Machine and Installation

Even after the size of motor has been checked up by the requirements discussed, it is important to consider the type of machine and the general class of installation. For example, suppose the motor drives a centrifugal pump elevating water to a certain height. A variation in head or speed will greatly modify any test figures given; therefore, allowance should be made and the motor installed of sufficient size to take care of such contingencies. If the installation is in a very dirty place a squirrel-cage motor would evidently be preferable to any other type of induction motor. If the voltage is apt to vary this must be taken into account, particularly with induction motors in which the torque varies as the square of the voltage. If the motor is installed in a hot place it should be larger than if installed under normal conditions.

Alternating-current Motors

By far the greater number of alternating-current motors are of the multiphase-induction type, the construction of which is too well known to need describing in detail. Briefly, these motors consist of a stationary element, or stator, to which the line current is supplied, and a revolving element or rotor, not connected electrically to the stator but which is acted upon by the flux induced by the current in the stator and is caused to rotate at some fixed speed dependent on the number of poles and frequency of the circuit. The great advantage which the multiphase-induction motor possesses over any other type is its simplicity.

There are two general types of constant-speed induction motors—the squirrel-cage and the wound-rotor types. The former is so called because the conductors in the rotor consist of copper bars laid in slots and short-circuited at the ends by metal rings; hence the resemblance to a cage. The rotor conductors in the wound-rotor type are form wound, somewhat similar in appearance to the armature coils of direct-current machines or of alternators, and these coils are connected to collector rings on the shaft. These collector rings are similar to those on alternators and are supplied with brushes from which the rotor currents are led through some sort of controller and suitable resistance. This resistance may be designed for starting duty to remain in circuit only while the motor is coming up to speed, or it may be constructed to remain in circuit continuously, where a reduction in the speed of the motor is desired. It should be noted that squirrel-cage motors are usually constant speed, whereas wound-rotor motors may be either constant or variable speed.

In considering the choice of a suitable constant-speed induction motor for any given installation the standard squirrel-cage motor offers the advantages of simplicity and ruggedness, with no moving parts carrying current; hence a motor particularly suitable for dirty and dusty locations or where sparking might be dangerous. The disadvantage of this type is that the starting torque per ampere of current is low. To develop a starting torque equivalent to full load torque the motor will draw from the line momentarily a current several times that necessary at full load. This is apt to cause considerable disturbance in the line and affect other apparatus connected to it, particularly if the motor is large. If lights are connected to the same line they may flicker badly. All squirrel-cage motors, except very small ones, if started by throwing directly on the line, would take a heavy rush of current. To prevent this a suitable form of starting device, or autotransformer is employed to reduce the impressed voltage when starting. The starting torque, however, is proportional to the square of the impressed voltage, so that it will fall off faster than the current. Hence,

if the motor is to develop a large torque when starting, it will take a comparatively large current from the line at that time.

The maximum torque of induction motors is generally from two to three times full load torque. This maximum torque is independent of the rotor resistance and, therefore, with any given motor, the maximum torque which it is capable of exerting under given conditions will be the same regardless of whether the rotor is of the squirrel-cage type or of the wound-rotor type or any modification of them. So far, therefore, as maximum torque is concerned a motor can be selected with regard to its starting torque only, knowing that the type of motor will have no effect on the maximum torque which is to be developed.

The wound-rotor motor is particularly suitable where the starting duty is severe, where a large current drawn from the line at starting is objectionable, and the installation is in a place free from dust and dirt. The starting current, corresponding to full load torque, is only a little greater than full load current as against four or five times as much in the case of the standard squirrel-cage motor as already noted. The maximum torque is exactly the same for the same horsepower and speed rating. The efficiency of the motor is lower, due to increased losses in the rotor and the drop in speed from no load to full load, or the slip, is greater than in the standard squirrel-cage type.

Where a heavy starting torque is desired and the wound-rotor type is not desirable a squirrel-cage motor with a high resistance rotor may be used. This is similar to the ordinary squirrel-cage motor, except that instead of the low resistance rings connecting the ends of the rotor bars, these end rings are of less cross-section or of a different metal so that their resistance is greatly increased. Since in the commercial induction motor, other things being equal, the starting torque is proportional to the rotor resistance up to a certain point, the construction just described results in a greatly increased starting torque, though the efficiency is lessened and the slip from no load to full load is greater. This type requires very careful construction of the end rings so that the heat developed by the heavy currents in the rotor at the time of starting will not loosen the connections and therefore cause poor contacts.

A significant point inherent in the design of induction motors is that, as the speed decreases, the operating characteristics become poorer. As the speed decreases for a given horsepower rating the magnetizing current of the motor increases so that the power factor is much less for the lower speeds. The efficiency is also decreased, though not as rapidly as the power factor. With a standard squirrel-cage motor the starting torque is also affected, being lower on the slower speeds. Hence, wherever possible, motors of as high speed as possible should be used, not only on account of first cost, but because of better operating characteristics.

From the standpoint of first cost the standard squirrel-cage induction motor is cheapest, the squirrel-cage motor with high resistance rotor next, and the wound-rotor motor is highest.

The use of variable-speed, synchronous, single-phase and direct-current motors will be considered in subsequent articles.

Improving Conductivity of Joints

It is obvious that considerable loss may occur through imperfect metal-to-metal contact on switchboards, especially where the copper is bolted together and the pressure is not very great.

A test made recently at the Laboratoire d'Electricite, on joints made in accordance with the best French practice, showed that on a busbar carrying 1760 amp., 84.5 watts were lost per hour due to heating. In a plant operating 10 hr. per day and 300 days per year, this would represent 253 kw.-hr. per year. While this may not appear large, it should be remembered that this was only one of a number of similar joints on the switchboard.

A preparation has recently been put on the market which is claimed to reduce this loss by about 85 per cent. It is called "Optimus Plastic Alloy," and is in the form of a metal putty, which is a good conductor and fills up all the cavities in the surface; thus, it is claimed the plates are made continuous electrically. It is further claimed that moisture is prevented from entering.

The alloy is put in cold and does not prevent the copper from being separated readily when desired.

Rotary Converter without Shunt Field

I should like to have the opinions of some readers as to what would occur if a loaded rotary converter should suddenly loose its shunt field. What proportion of full load could the machine carry under these circumstances and what would be the probable power factor?

Schenectady, N. Y. N. M. Hill.

Peculiar Motor Trouble

A short time ago upon placing a 10-hp. induction motor in service the following trouble was encountered:

When the compensator was on the starting position the motor, with an ordinary light load of shafting and loose pulleys, would develop about one-third normal speed and with a consequent lack of power. This would not change when the compensator was brought to the running position. If the handle of the starting compensator was thrown quickly from the "off" to the "running" position, the motor would reach almost the rated speed, but the moment the load was thrown on, it would slow down to about three-fourths normal and heat considerably to about 185 deg. F. or 115 deg. above room temperature.

This motor drove two triplex ammonia pumps and two triplex brine pumps, which load is at present being carried by a motor of the same type and size, which shows that it was not an excessive load that caused the trouble.

The motor in question is a three-phase, 60-cycle, 550-volt machine of the squirrel-cage type and is rated to run at 1200 r.p.m. The stator winding consists of 72 coils, arranged in 18 groups of four coils each. The stator winding had just been renewed and the cross-connections were made in exact accordance with the manufacturer's drawing for Y-wound motors of this type. A voltmeter test indicates a ground of 20 volts, but I do not think that this would cause such conditions to exist.

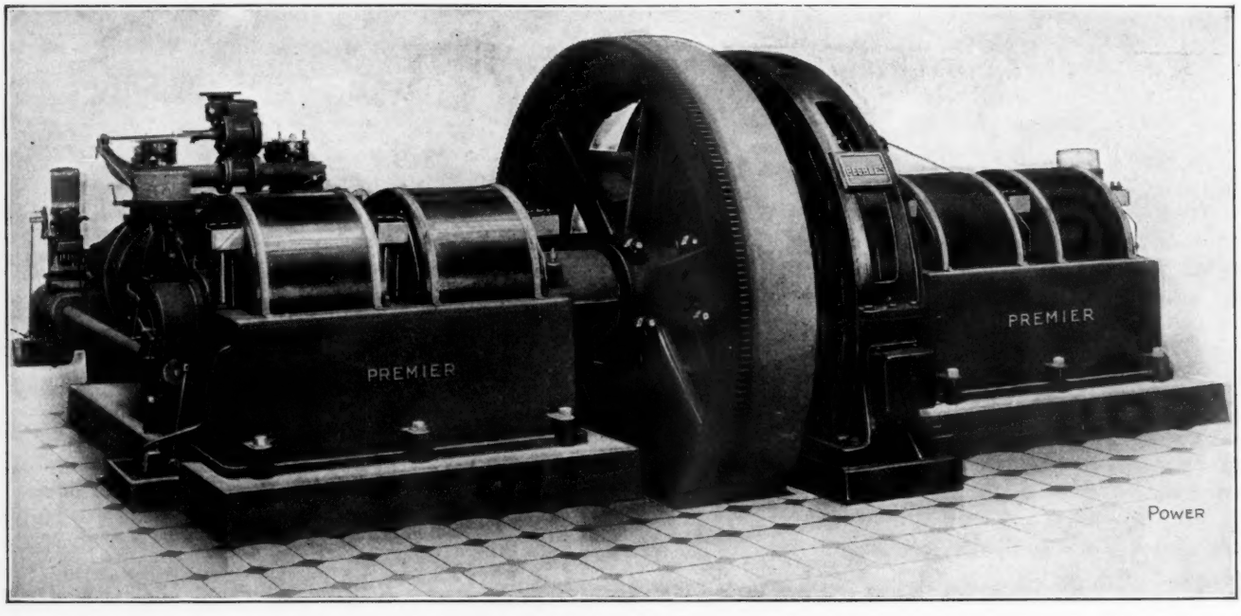
I have examined the rotor for loose bars and it is in good condition, as was the starting compensator, nor do the bearings show any signs of wear.

Washburn, Wis. F. L. Bryant.

GAS POWER DEPARTMENT

A Korean Gas Engine Plant

The application of gas power to industrial activities is finding its way into all parts of the world and, in this connection, it is astonishing with what avidity the newer nations are adopting the most advanced forms of engineering practice which have been developed in the older world. An interesting example of this is found in Japan and Korea, where a number of townships which comparatively recently were highly primitive, are rapidly adapting themselves to the conditions of modern civilization. In many of these, efficient systems of gas and electricity supply are to be found and some of the generating stations or power houses are models of their kind.



ONE OF THE MAIN GENERATING UNITS

The Kankoku Gas & Electric Co., in Korea, is a flourishing and progressive company supplying gas and electricity to the city of Kankoku. Its power-house equipment consists of two double Premier gas engines, each direct connected to a 300-kw. alternator. The alternator and flywheel are placed centrally between the two pairs of cylinders of the double twin gas engine, which has four single-acting cylinders working on the four-stroke cycle and there are therefore two impulses per revolution. The illustration shows one of these units.

At present the engines operate on town gas, but they are so arranged that they can be adapted to work on producer gas, should this become necessary. The quality method of governing is employed, a high-speed centrifugal governor driven from the crankshaft controlling the throttle valve for the gas. The gas and air do not mix until about to enter the cylinder. Ignition is obtained by a separate magneto for each machine.

Starting is effected by compressed air, a distributing valve, operated by a cam on the camshaft, supplying air at the right point in the cycle through a check valve opening into the two cylinders. This gear can be readily put into or out of operation at any position and it continues to admit air every cycle until ignition commences, when the admission of air ceases.

Lubrication of the main bearings and crank pins is effected by a valveless oil pump delivering oil to these parts, and drawing it from a tank connected to the crank chamber, after passing through a strainer. The cylinders and exhaust valves are lubricated by a sight-feed arrangement and a small force pump. The small-end bearings of the connecting-rod are supplied in a similar way. This lubrication gives a constant feed after it is once adjusted, which stops when the engine stops and starts when the engine starts. The measured oil drops into the suction side of a small pump which delivers it under pressure to the point where it is required. The gear wheels work in oil-tight cases and in an oil bath and the shaft bearings in these cases are lubricated by the oil thrown up by the wheels, thus keeping a constant stream flowing over these bearings. The other camshaft bearings are of the self-oiling type.

⁂

The Gas Producer and Waste Fuels*

The steam engineer aims at minimum CO and maximum CO_2, while the producer engineer strives for maximum CO and minimum CO_2. A boiler works with a fuel bed usually varying in thickness from 3 to 12 in., whereas the depth of fuel bed in the gas producer varies from

*Excerpts from a paper by F. Fielden, read before the Liverpool Engineering Society, Jan. 8, 1913.

2 to 10 ft. Maximum temperature in the furnace is the ambition of the fireman; on the other hand, a combustion zone of approximately 2000 deg. F., but varying with the nature of the fuel, gives the best results in the gas producer.

In boiler firing it is comparatively easy to deal with clinker, and the tarry volatile gases given off during the coking of the coal are never the cause of trouble at the steam engine and turbine. Stated broadly, however, clinker and tar, are a source of the greatest trouble to the designer of the gas producer.

The earliest application of the gas producer was confined exculsively to heating installations, and only with the advent of the Otto cycle gas engine was begun the adaption of the gas producer to the problem of cheap power production.

The bases of gas production depend on the reactions taking place principally in accordance with the following equations:

$$2\ C + O_2 = 2\ CO$$

(or $C + O_2 = CO_2$ followed by $CO_2 + C = 2\ CO$)

and

$$C + H_2O = CO + H_2$$

If the capacity and design of the suction producer is suitable for the engine, the foregoing reactions are governed entirely and automatically by the load. Since there is no outlet for the gas from the generator except to the engine, it follows that the induction stroke of the engine determines the amount of air and vapor drawn through the fuel bed and consequently the amount of gas produced.

The steam, or vapor supply, has a twofold effect; first, in preventing an abnormal temperature at which the ash would be fused and the firebrick lining burned, and secondly by dissociation, enriching the gas with hydrogen. This steam is generally obtained from the sensible heat of the gas in its transit to the cooling and cleaning scrubber.

Since it is necessary in either a steam boiler or a gas-producer installation to make allowances for fuel consumption during standby hours, the consumption at full load is not the only factor. The standby consumption per hour with a well designed suction-gas producer plant is about 2 per cent. of that at full load. With a steam-boiler installation the standby consumption per hour is nearly 15 per cent. of that at full load.

In the classification of coal, bituminous coal comes between lignites, at the lower end of the scale, and anthracite at the upper end. A general classification as to volatile matter is as follows:

Anthracite up to 8 per cent. volatile matter.

Semi-anthracite, 8 to 10 per cent. volatile matter.

Dry steam coal (navy), 10 to 14 per cent. volatile matter.

Semibituminous, 14 to 20 per cent. volatile matter.

Bituminous, 20 to 40 per cent. volatile matter.

To the gas-producer designer, difficulties are intensified. For power purposes the gas must be clean and free from tar, and of consistent quality. Hot burning coal must be fed with a liberal supply of steam to reduce ash fusion and clinker formation; also, arching over of caking coal must be prevented and a close fire of sufficient density maintained for the reduction zone.

In the same generator it is practically impossible to gasify with commercial success every grade of bituminous coal. An area of combustion zone giving a combustion of 20 lb. to the square foot is ample for Nottingham slack, whereas for Denver slack, double this area is required.

The assistance of the analytical laboratory is necessary to the designer in the adaption of the producer to the requirements of bituminous coal. The chief stumbling block, however, is tar trouble, with its accompanying effect on the working of the gas engine. Sticky valves and piston rings and throttle ports are the direct cause of irregular working and ultimate stoppage of the engine. Many attempts have been made to fix the tar in the form of an incondensible gas in the producer itself by employing down draft or up and down draft in which the products of distillation are made to pass through an incandescent mass of carbon, but up to the present only partial success has been attained. Some firms prefer to extract the tar from the gas externally by means of a centrifugal tar extractor.

During the past five or six years the bituminous suction plant has been successfully applied to the gasification of waste fuels, and pioneer installations are at work dealing with sawdust, wood refuse, spent tan, cocoanut shells, coir dust, cotton seed cake, coffee and cocoa husks, smoke-box char, olive refuse, etc. While some of these fuels have been employed for boiler firing, the consumption per brake horsepower-hour is two to four times that of the suction-gas plant. On the other hand, it has been found necessary to use coal with some moist fuel, such as spent tan and wood refuse, in order to get the necessary temperature in the boiler furnace. A moisture content of 50 per cent. appears to be no detriment to gasification in the correct design of producer.

Such fuels as smoke-box char, weighing 50 lb. per cu.ft., demand a different design than that for such fuels as fine sawdust or husks. The former demand an excessive area of cross-section with a minimum depth of fuel bed, and an average velocity of gas when leaving the top of the fuel bed. On the other hand, the sawdust and light fuels require a maximum depth of fuel and a minimum velocity of the gas as it leaves the fuel bed, to prevent pulling over the raw fuel during the induction stroke of the engine.

The essential difference in the relative combustion of light, moist fuel in a gas producer as compared with a steam boiler seems to lie in the fact that with the former a sufficient temperature at the base of the fuel bed is necessary, together with an adequate density of incandescent carbon and the requisite velocity of the air and vapor through the fuel bed to effect reduction; while with the steam boiler radiant heat is what is most desired. To obtain this it is necessary to drive off all moisture which at once extracts heat from the boiler, and at that particular stage of the process when the top layer of fuel is in a combustible state the draft is sufficient to draw the carbonized particles away along the flues and even up the stack.

That the gas producer is one of the most commercially profitable means for the utilization of waste fuels is shown by the large number of such installations now in operation. At the Highfield tannery, at Roencorn, an installation of 100 hp. has been working for some years on spent tan alone, with such satisfaction as to warrant the recent duplication of this unit. A more recent installation, the largest in the world for dealing with this

waste product, is one of 700 hp. now being erected at Ditton, in which 300 hp. is to be used for power purposes, and 400 hp. for combustion under existing steam boilers. With a moisture content of 50 per cent., the consumption per brake horsepower-hour amounts to approximately 3½ lb., and the gas produced from this weight of spent tan is from 70 to 80 cu.ft., at approximately 125 to 135 B.t.u. per cu.ft.

In a recent test of a 75-hp. wood producer plant, employing chippings and wood waste as fuel, a brake horsepower-hour was obtained with 2.64 lb. of refuse. This fuel had a heat value of 163 B.t.u. per lb., and the engine at the time was running at 15 per cent. overload.

Test on Diesel Engine Using Tar Oil

A 325-hp. vertical, single-acting, four-stroke-cycle Diesel engine running 167 r.p.m. in the Hamburg-American Watch Factory, at Scharmberg, Germany, was recently tested by the Wurtembergische Revisions Verein in Stuttgart. The engine is direct connected to a three-phase generator, the two single-stage air compressors being driven directly from the shaft. A series of tests at one-fourth, one-half, three-fourths and full load, as well as at 10 per cent. overload was made to ascertain the fuel consumption, including the gas oil, which is used in starting. The results of these tests are graphically shown in the accompanying diagrams. The highest output attained was 240 kw., or 355 effective horsepower from the engine, which is 10 per cent. more than was guaranteed.

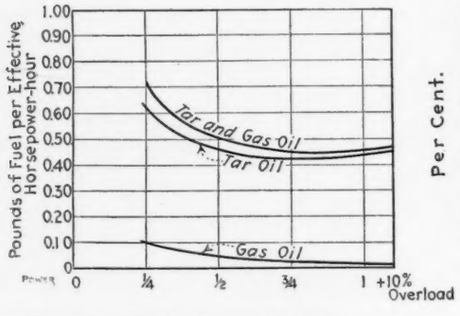


FIG. 1. FUEL CONSUMPTION VS. LOAD

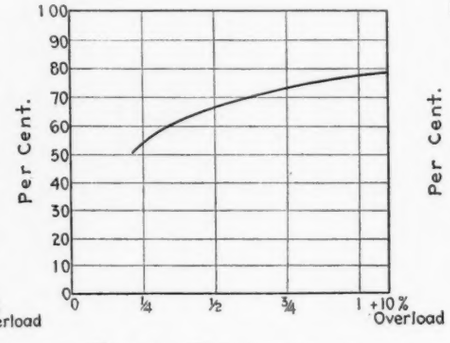


FIG. 2. EFFICIENCY VS. LOAD

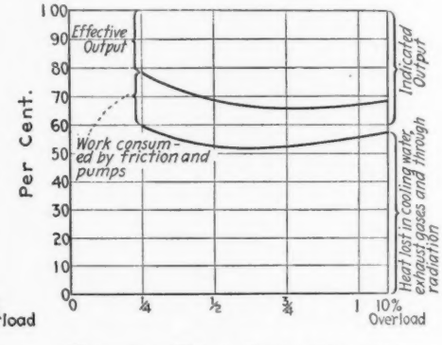


FIG. 3. HEAT BALANCE WITH DIFFERENT LOADS

With fuel prices such as obtain in Germany the cost per kilowatt-hour figures 0.35c. per kw.-hr., or 0.23c. per effective horsepower.

Fig. 1 shows that the most favorable fuel consumption occurs between three-fourths and full load and remains nearly constant, which speaks well for the operating economy of the Diesel engine. Fig. 2 shows that the mechanical efficiency at full load is about 77 per cent., and at 10 per cent. overload about 78.5 per cent. These figures reflect favorably on the construction and lubrication.

Fig. 3 is a composite plot of the heat balance for various loads. It shows that about 55 per cent. of the heat contained in the tar-oil fuel is lost in cooling water, exhaust gases and through radiation, while at full load about 34 per cent. is gained as useful work, the rest being absorbed by the pumps. This diagram also shows that the most favorable operating conditions are obtained between three-fourths and full load.

CORRESPONDENCE

Drying Out Magnetos

The company with which I am employed operates a number of launches, having magnetos of the three-bar permanent magnet type for ignition purposes. Not long ago the magnetos began to give trouble, which upon test, proved to be due to salt-water grounds in the armature. As the magnetos are friction driven from the flywheels of the engines, they were installed at about the same level as the engine base, which made them subject to all the dampness and salt water that accumulated in the bottom of the boats.

The trouble was remedied in the following manner: The pole-pieces were cleaned and shellacked; then the armatures were put in a pot of boiling water and allowed to boil for about 20 min., after which they were baked in a crude, but successful, manner by suspending over the stationery boilers for 24 hr. They were then shellacked thoroughly and allowed to bake another 24 hr., after which they were assembled, and have given no further trouble.

Oakland, Calif. J. T. BROWN.

Pound in Engine Cylinder

Replying to the inquiry of Mr. Able, in the Jan. 28 issue, as to the possible cause of the pound in his oil engine, I would suggest that he examine his engine for carbon deposits on the cylinder head and piston.

I have had experience with cases similar to his, particularly with engines using low-grade fuels, and the cause was usually carbon deposits; although, in one instance, it was a burr on the cylinder head which, when heated, caused preignition. This gave no trouble until the engine had been in service some time, as the cooling water kept the burr below the ignition temperature, until the water jacket became badly scaled from the cooling water.

In either event, the engine would act the same, for the mixture being ignited by the incandescent carbon or the burr, advancing or retarding the spark would have no effect. At light load, the compression would not be high enough to cause ignition to occur early enough to cause a pound and, when using too rich a mixture, the combustion would be too slow and weak to cause a pound.

Rush City, Minn. C. W. FOSTER.

REFRIGERATION DEPARTMENT

A New Refrigerating Cycle

By Oscar P. Ostergren

Carbonic-acid machines, when operated as at present, have shown surprisingly good practical efficiency—a fact that should encourage further efforts toward improvement. The ratio of compression is not so great as in the ammonia machine, which is an important consideration. The losses at the regulating valve, however, must be greater. So, also, is the loss due to the mechanical efficiency of the machine on account of the unusually high pressure required. To lower this pressure is the object

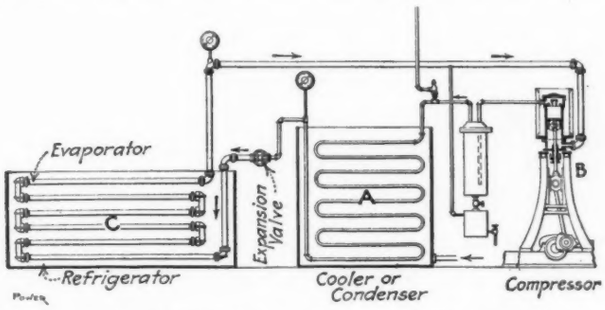


Fig. 1. Connections of Ordinary Compression System

of the proposed cycle. Fig. 1 shows the usual compression system and Fig. 2 the new cycle.

Referring to Fig. 2, carbonic anhydride in a gaseous state enters compressor *A* through suction valve *a*, where, after compression, it is discharged through valve *b* into oil trap *c* and cooling coil *d*, where it is cooled by water in the regular way. From *d* the gas enters the countercurrent cooling apparatus *B*, which may be termed a "precooler," and is then conveyed by the inner pipe *e*, through *B*, to a second and similar apparatus *C*, which will be called a liquifier. The gas passes through the inner pipe *f* to the expansion valve *g*. In pipe *f* the gas or vapor is condensed into the liquid state at a comparatively low temperature. In passing the expansion valve *g*, under a reduction of pressure, part of the liquid is evaporated to reduce it to the temperature existing in part of the liquifier. From the valve, the mixture of liquid and vapor enters the exterior pipe *k* and proceeds to the coils of the refrigerator *D*. In passing through the liquifier, heat is taken up by the cold mixture from the compressed gas in pipe *f*, which runs in the direction of the expansion valve *g*, thereby liquifying the gas. The transfer of heat in the liquifier should be carried to the extent that the CO_2, when entering the refrigerator, is at the desired temperature. This can be regulated as desired by means of by pass valves *m* and *n*. If a lower temperature is needed, *m* is opened and also *n*. Less gas will pass through the refrigerating coil *D*, but it will be of lower temperature. Having passed through the refrigerator, where heat has been taken up from the body to be refrigerated, the refrigerant now enters the exterior pipe *o* of the precooler, where still more heat is taken up from the refrigerant, which is going toward the liquifier. From the precooler, it returns to the compressor at nearly the temperature of the cooling water. In this system it will be possible to get any desired temperature in the refrigerator between that of the cooling water and a point somewhat above the solidifying or freezing point of the refrigerant without any other change in the arrangement than some adjustment of the expansion valve *g* and the bypass valves *m* and *n*. With this regenerating process, the refrigerant can be liquified at any practical pressure. The back pressure (low pressure) may be anything desired. The process is also practical when the cooling water is of relatively high temperature and should, on this account, be of special value in hot countries. The system should lend itself particularly to small refrigerating plants and also to special low temperatures. It is practically out of the question to operate a very small compressor continuously against a pressure of over 1000 lb.

While the above process is, so far as the author knows, original with him, it is not entirely new. Several years ago, he had the opportunity to carry on research work on an extended scale in connection with low temperatures,

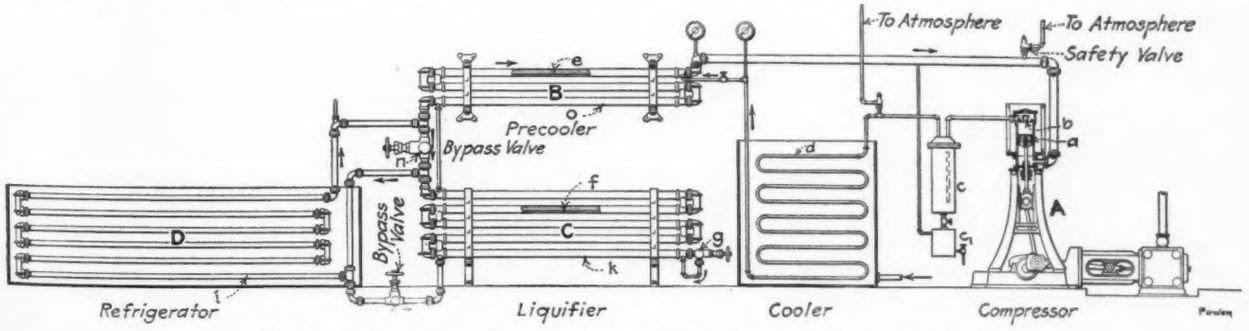


Fig. 2. Diagram of Apparatus for New Refrigerating Cycle

where large quantities of many of the so called permanent gases were liquified, including atmospheric air. Some compound gases, such as the air, illuminating gas and even gases from combustion in an ordinary furnace, were cleaned and separated into their component parts by means of partial liquification. Even 15 years ago, it was

quite plain to the writer and others connected with this kind of work that great improvement in the efficiency of liquification cycles could be obtained by precooling the liquid refrigerant before expansion. In this way, it was possible to liquify air at a pressure of about 800 lb. or even less; whereas 3500 lb. had been used. Without the regenerating process, it would have been necessary to use a pressure of 10,000 to 12,000 lb. per sq.in.

Carbonic-acid refrigerating machines are taking a prominent part in the refrigeration of ships today, supplanting both the ammonia and dense-air machines. There seems to be no reason why their use should not be extended to railroad refrigeration on a large scale. They possess many qualities which the ammonia machine lacks and, with some further improvement in their thermal efficiency, and the reduction of the working pressures to practical limits, they should prove to be a formidable, if not superior, rival of the latter.

[The new refrigerating cycle described in this article is intended to be based on the construction of the machines used for manufacturing liquid air, and while such a system for practical refrigeration would require a great deal of power per refrigerating unit, it is here presented, as the suggestion on reducing the operating pressure may be worthy of some consideration.—EDITOR.]

Avoidable Accidents in Refrigerating Plants*

BY LOUIS BLOCK

It is the custom to test ammonia piping with air pressures of from 250 to 300 lb. per sq.in. before charging the system with ammonia. The inexperienced operator is apt to meet with an explosion, when the air temperature reaches 500 deg., and this may occur as soon as the pressure has reached 110 lb. The usual mode of procedure is to lubricate the cylinders in the ordinary manner with mineral oil, which is thereafter to be used in the refrigerating system. This oil is eventually discharged by the compressor into the piping, where it lodges in some pocket, and finally when a sufficient high temperature reaches this pocket, is vaporized and ignited.

I advise covering the cylinder walls with a thin coating of lard oil, using the hand to apply it; then compressing up to 100 lb. and stopping long enough to allow the compressor to cool; then running up to 150 lb., and again stopping to let the compressor cool. After this the compressor should be operated at a slow speed and stopped as soon as the discharge pipe becomes so hot as to make it impossible to keep one's hand on it for a period of a minute.

A number of operating engineers have the habit of pouring hot water over the ammonia shipping drums to facilitate the charging of the ammonia into the system and some that have a steam jet handy let steam impinge against the drum, or they sometimes wrap cloth around it and stick the end of the steam hose under the cloth. If the engineer turns off the valve on the pipe connection between the shipping drum and the charging pipe and goes to lunch, forgetting to turn off the steam, the drum will probably explode.

*Abstract of paper read before the American Society of Refrigerating Engineers, New York City.

Ammonia gas from leaks, due to the breaking of a pipe, flange or fitting, when there are lights or any other open flame present, may ignite and fill the entire engine room for a few seconds with fire. Such accidents can positively be prevented by having nothing but incandescent lamps in the engine room.

The breaking of compressors may be due either to leaving the stop valve in the discharge pipe shut when starting up, or to dropping a valve into the compressor. The former class of accident may be prevented by installing the usual bypass and safety valve. To prevent the latter class of accident, use valves having hollow stems wherever possible. Anneal the valves every year and throw them away after they have been in use five years.

The breaking of crossheads in steam engines is usually due to water in the cylinder and may be prevented by superheating the steam or installing an efficient steam separator close to the engine.

The following rules may be worth remembering:

Don't use a compressor which has been used for ammonia compression, for compressing air.

Don't calk a fitting while under ammonia pressure.

Don't stand in front of a steam cylinder or compressor unless it is absolutely necessary.

Rating a Compressor

The French Association of Refrigeration, in the *Bulletin* of the International Association, expresses the wish that the following definitions should be adopted:

1. The normal power of a refrigerating machine is the number of frigories* it can absorb in one hour for gas temperatures of 77 deg. F. at the condenser, 59 deg. F. at the expansion valve and 14 deg. F. at the refrigerator.

2. The specific volumetric production is the number of frigories really absorbed at the refrigerator by each cubic meter of gas drawn in by the compressor.

3. The volumetric efficiency is the relation between the number of frigories really produced by the suction of the unit of volume of vapor and the number corresponding to the cycle of Carnot.

4. The specific economical production by the number of frigories produced per indicated horsepower of the compressor.

5. The economical yielding by the report of the specific economical production similar to that of a machine realizing the cycle of Carnot.

*One frigory per hour equals 0.00033 American tons per 24 hr.

Ice Machine Racing

In reference to the inquiry on this point, appearing in the Dec. 24 issue, it has been my experience on a 150-ton compressor that after a certain speed, which will vary a little with different suction pressures, enough gas will not be obtained through the suction valves to fill the cylinder to its full capacity. The cylinder not being filled to its capacity, there is less gas to compress and less work to do, so that the engine will tend to run away. I have had an engine race badly by speeding up an overloaded machine in an attempt to get more work out of it.

Revere, Mass. R. B. ROWE.

EDITORIALS

Caring for Condenser Apparatus

Many engineers carry too high a vacuum in condensers serving reciprocating engines and too low a vacuum in condensers serving turbines.

Because, under usual conditions, it is difficult to maintain a satisfactory degree of vacuum without paying careful and constant attention to the condensing apparatus, some think that the higher the vacuum maintained the better it speaks for the engineer in charge. Theory shows, and practice proves, that, with reciprocating engines operating under everyday conditions, there is a decided gain in the economy or water rate of the condensing engine up to a vacuum of between 26 and 27 inches. Tests of a 5500-horsepower engine at the Waterside station of the New York Edison Co., made during 1904, showed that when the vacuum was increased from 25.3 to 27.3 inches, the water rate decreased only 0.06 pound per indicated horsepower. The tests and the results may be considered typical of what may be expected of any reciprocating condensing engine. The operating engineer should remember that, with degrees of vacuum of over 27.5, or at most 28 inches, the increased loss due to cylinder condensation goes far toward nullifying the gain due to a better vacuum. Cylinder condensation is caused by the alternate heating and cooling of the cylinder walls at each stroke of the engine. On the exhaust stroke, the cylinder is, of course, cooled considerably and, when live steam is admitted and encounters comparatively cold cylinder walls, it gives up a part of its heat, which, neglecting the small amount restored by reevaporation during expansion, is lost as far as doing useful work is concerned.

As we all know, the condensate should be returned to the boilers at as high a temperature as possible consistent with the best degree of vacuum. The higher the vacuum carried, the lower will be the temperature of the condensate or hotwell water. For example, at 26 inches vacuum the temperature will be 125.38 degrees Fahrenheit, at 27 inches 115.06 degrees, or a drop of 10.32 degrees. The drop in temperature from 27 to 28 inches vacuum is 13.91 degrees and from 28 to 29 inches it is 22.08 degrees. Knowing that ordinarily every ten degrees rise in the temperature of the feed water decreases the coal consumption one per cent. we realize the importance of not maintaining too high a vacuum. The temperature of the hotwell water will nearly always be below the figures given above on account of water leaking through the tubes and of other causes.

With turbines, cylinder or initial condensation is practically nil, as the high- and low-pressure ends of the machine are at a constant temperature as long as the initial steam pressure and the vacuum are kept constant. Obviously, then, there is no limit to the degree of vacuum that may be economically carried on a turbine.

After all, a condenser may be considered as a heat exchanger, wherein water, by direct or indirect contact with steam, absorbs heat from it. The more perfect the heat absorption, or exchange, the more perfect the vacuum.

In condenser practice, the chief obstacle to creating and maintaining a high vacuum is the presence of air in the condenser. To insulate a steam pipe, the aim is to surround the pipe with a film made of cells of dry "dead" air. Because air is an excellent thermal insulator, it is as detrimental in a condenser as it is good in a steam-pipe covering.

As the steam condenses in a surface condenser, much of the air in the steam is liberated and left behind. As the process continues, the density of the air surrounding the tubes increases, forming an insulator for the tubes —the density increasing as the bottom rows of tubes are approached.

While the purpose of the air pump is to remove air from the condenser, the engineer should make sure that it is not removing air from the condenser that could be kept out of it by carefully guarding against leaks. When the air pump is worked unnecessarily hard, it means wear and tear of the pump and a higher steam consumption, which is serious in any condensing plant.

In most condensing plants, much of the air gets into the condenser through small but numerous leaks in the condenser shell and piping. The feed water is also a conveyor of air, although it is claimed that where open feed-water heaters are used the air going to the condenser with the steam is greatly reduced.

The sealed glands of the turbine and air pumps admit air if the seal pressure is below normal. The results of tests made to determine the amount of leakage at the seals, were brought out in a discussion of a paper by George A. Orrok, on "Air in Surface Condensation," recently presented to the American Society of Mechanical Engineers.

In the installation tested, the condensers were of the jet type, serving a 2000-kilowatt turbine. At no load, and with no steam on the seals, 86 cubic feet of free air per minute entered the condenser; a corrected vacuum of 26.6 inches was maintained. At about 80 per cent. rated load, and with no pressure on the seals, 91 cubic feet of free air per minute leaked into the condenser, causing the vacuum to fall. With normal pressure on the glands, the leakage fell to 58 cubic feet and the vacuum rose to over 28 inches.

To get an idea of the amount of air leaking into large condensing turbines, Mr. Orrok's accounts of tests in this direction are indeed valuable and interesting.

For units between 5000 and 20,000 kilowatts, the amount of air discharged by the vacuum pumps varies from one cubic foot per minute (free air), when the units are in the best condition, to 15 or 20 cubic feet, when ordinary leaks occur. When the units are in bad condition, the air handled by the air pumps varies from 30 to 50 cubic feet per minute. The above figures show how necessary it is to guard against air leaks.

There is also too great a tendency for engineers not to allow sufficient temperature difference between the inlet and outlet circulating water. The aim should be to make the water absorb as much heat as possible, consistent with maintaining a high vacuum. When this is not done, the steam consumption and the deterioration of the circulating pump are necessarily increased.

Steam, inlet and outlet circulating-water temperatures, time and vacuum readings, taken at short intervals and recorded on a proper log sheet, are about the best means available for knowing how well the condensing apparatus is operating.

Another thing the engineer should bear in mind is this: The cost per kilowatt of high-vacuum apparatus increases much more rapidly than the increase in vacuum maintained by the apparatus.

After Coal—What?

The yearly production of vegetable matter on the earth is estimated at 32,000,000,000 tons, which if burned would yield a quantity of heat equal to that obtainable from 18,000,000,000 tons of coal—that is to say, eighteen times as much heat as is furnished by all the coal now mined in Europe and America.

Is it not to be supposed that this production of plant substances may be so increased and intensified in selected regions as to supply all the fuel we can possibly want in the future? We are now using for fuel purposes fossil solar energy in the form of coal. Why not utilize instead live solar energy?

Plants should be made to store up solar energy for conversion into mechanical energy. The kind of plants used is of no consequence. They may be grasses or trees; they may grow in swamps or dry places, on the seacoast or even in the sea. The essential point is that they shall grow fast or that their growth shall be accelerated.

The production of plant substances over all the land surfaces of the earth averages one ton to the acre. By intensified culture it could be made four tons. This on one square mile would amount to 2560 tons, corresponding to about 1400 tons of coal.

In an epoch soon to arrive the harvest of fuel plants dried by the sun may be converted entirely into gas for fuel purposes, separating out the ammonia, to be returned to the soil as nitrogenous fertilizer, together with all the mineral substances contained in the ashes. The gas thus obtained may be burned on the spot in gas engines or otherwise and the mechanical energy generated by this means be transmitted over great distances or utilized in any way that seems advisable. Even the carbonic acid resulting from combustion could be returned to the fields, so that the latter may lose nothing of their productiveness.

By this method the radiant energy of the sun may be made to furnish low-priced mechanical energy, plants (instead of the mirrors with which promising experiments are now being made in Egypt and Peru) being used as accumulators.

Few people have any notion of the vast quantity of live solar energy available for use. In the tropics with a day of six hours' sunshine, the sun delivers upon one square mile per day an amount of heat equal to that obtainable from the burning of 2500 tons of coal. A surface of 10,000 square miles receives in a year at this rate a quantity of heat equivalent to the burning of 9,000,000,000 tons of coal—which is nine times the total amount of coal mined in Europe and America in a twelve-month. The Desert of Sahara, with an area of 2,280,000 square miles, receives daily solar energy equal to 6,000,000,000 tons of coal.

Compared with this live solar energy the quantity stored up by plants in ancient geologic epochs in the form of coal is trifling.

Recent discoveries of science have proved that sunlight itself may be utilized to develop immense energy. Accordingly it is suggested by Professor Ciamician that desert regions of the tropics, where conditions of soil and climate render it impossible to grow crops, shall be made to employ the sun's rays through the medium of photoelectric batteries, or even photochemical batteries, delivering energy equal to that of billions of tons of coal.

Solar energy is not evenly distributed over the surface of the earth. There are privileged regions in the tropics which would become vastly prosperous if the sun's rays were suitably employed. Even now the strongest nations are rivaling one another in the conquest of the lands of the sun as if unconsciously looking to the future.

Where vegetation is profuse, solar energy may be utilized through plant growth for industrial purposes—that is to say, for the production of mechanical power. In desert regions unadapted to any kind of cultivation, photochemistry will artificially put the sun's rays to practical uses.

On arid lands will spring up industrial colonies without smoke and without smoke-stacks. Forests of glass tubes will extend over the plains.

Coal offers to mankind solar energy in its most concentrated form. Indeed, modern civilization may be said to be the daughter of coal. But the supply of coal is not inexhaustible and every ton of it taken out of the earth leaves just that much less for the future needs of the world. Hence it is that today we are looking about us so anxiously for some other source of fuel.

John Fritz

What may be accomplished by earnest endeavor is strikingly shown in the life of John Fritz, a sketch of which is given on page 284.

At the age of sixteen, with what today would be considered a very meager education, he entered upon his mechanical career. The production of iron by modern methods had just begun, steel was practically unheard of, and structural steel was unborn. To commercially produce steel required more knowledge than the chemistry of it; machines subject to exceptional strains and stresses, and the design of which was unusual were required. The Bessemer and Siemens-Martin processes were tantalizingly slow in practical development, and American firms were, therefore, handicapped in competing with foreign manufacturers. Mr. Fritz, more than anyone else, was responsible for the successful development of these important proceses in the great steel industry of this country.

Engineering and scientific societies the world over have honored him, and both they and the steel industry will feel the loss of one active for nearly a century in engineering progress.

READERS WITH SOMETHING TO SAY

Improper Boiler Repair

That improper boiler-repair work is sometimes done if the workmen are not carefully watched, is shown by the following:

One of the main 1¾-in. diagonal stays was found to be cracked half way through in the shank weld, as shown in Fig. 1.

The brace was taken out and another made, using the old one as a pattern. The stay had a thread cut on the end going through the front head and was held in place by a 4-in. washer, both on the inside and outside of the

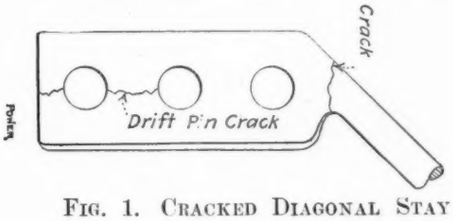


FIG. 1. CRACKED DIAGONAL STAY

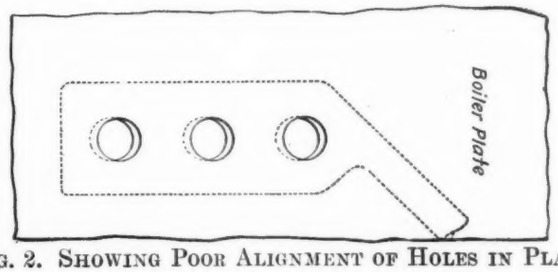


FIG. 2. SHOWING POOR ALIGNMENT OF HOLES IN PLATE AND STAY

front head. The nuts were set up solid, making the stays fast to the front head, while the rivet holes were marked on the end. Then the stay was taken out, the holes drilled and the stay put back. Noticing that the holes did not come even with the old ones, as shown in Fig. 2, by the dotted line, I spoke to the repair man about it. He laughed, saying that he would soon fix it. As I expected, he was going to "fix it" by using a drift pin and a heavy sledge hammer, but I insisted that he hammer the shank until it had increased in length $\frac{3}{16}$ in., making it long enough for the holes to register.

This incident illustrates the necessity of the plant engineer closely observing the repair work being done in his plant by outside men.

A. C. WALDRON.

Revere, Mass.

Value of Engine Inspection

The value of a systematic inspection of engines is frequently shown by the inspection reports. A recent inspection of a 15 and 30 by 18-in. high-speed compound-condensing engine revealed the fact that the high-pressure piston head had an opening about ½ in. in diameter through the side, and a little probing with a wire showed a much larger hole on the opposite crank-end side. The head was of the plain type without bull ring or follower plate, the chambers being cored out to lighten it. The piston was taken out, and a light tap with a hammer enlarged the hole to about 1½ in. in diameter. It was then seen that the walls of the piston were very thin, and lying at the bottom of the chamber was a piece of steel about ⅜ in. thick, and 1¼ in. square.

An examination of the chamber showed the source from which the steel plate came. When making up the mold for this casting the foundryman had to support the core that is used to form the hollow chamber. To do this he used a small piece of metal, known as a chaplet. When the metal is poured it flows all around this chaplet, so that it becomes practically a part of the casting. In this case the piece of steel was much larger than it should have been, and did not unite with the cast iron. When the machinist removed the metal from the face of the piston he cut through the chaplet leaving the inside piece loose, which fell down inside the chamber, and by being so rattled due to the motion of the piston had finally made a hole through the side of the casting.

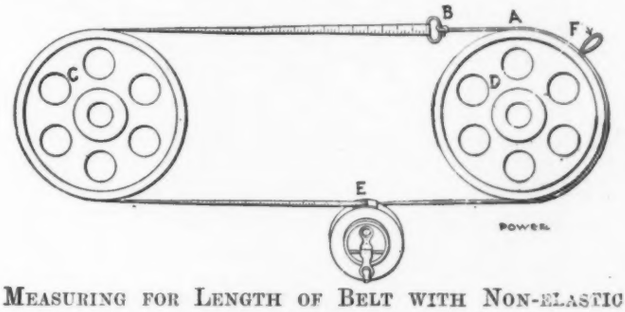
In this case, no great harm was done, only some steam lost, but if the chaplet had worn the hole large enough to allow the chaplet to get through and fall between the moving piston and the cylinder heads it would have undoubtedly caused trouble. The inspection in this case showed the defect before serious harm was done.

ARTHUR CLIFTON.

Joliet, Ill.

Belt Measuring Kink

Most engineers are familiar with the difficulties of accurately measuring the length of a belt with a string, especially when the shafting is in motion. Due to the



MEASURING FOR LENGTH OF BELT WITH NON-ELASTIC TAPE

amount of stretch (unknown) of the string, this method of measuring is a sort of hit-or-miss proposition.

These inaccuracies may be overcome by using a steel measuring tape, as shown in the illustration.

To measure, tie the string *A* to the ring *B*, which is on the end of the tape, pass the tape around the stationary pulley *C* and the string around the running pulley *D*, note the reading of the tape at *E*, add the string length

BE, and obtain the length of the belt. The knotted loop *F* is to facilitate holding the string on the running pulley. When measuring, the loop should be held between the fingers; it is dangerous to slip the fingers into the loop.

CHARLES HERRMAN.

New York City.

Placing a Large Suction Pipe

The accompanying illustration shows how a 48-in. suction pipe, 24 ft. long, of a large pumping engine was handled under difficult circumstances. It was made in two parts; the bottom piece was 10 ft. 9 in. long, bell-shaped at the bottom and flanged at the top. The upper section was 13 ft. 3 in. long and weighed 7 tons. As the building had been erected and the roof and other floors finished, it was necessary to pass those pipes through a hole in the foundation's wall. Some idea of the size and position of the pipe after entering the hole is shown in Fig. 1.

A platform was erected on top of the pump chambers to hold the hoisting crab, which was held in position by tying to a 2½-in. eye-bolt, screwed into the top of one of the adjoining pump chambers; also by passing a length of 2½-in. pipe through the openings in the crab and through two eye-bolts in the top of the chambers. As the suction well was only 12 ft. square, with smooth vertical walls, and an opening 4 ft. square in the center of the floor, it was necessary to place the lower half of pipe in a nearly correct position, so that there

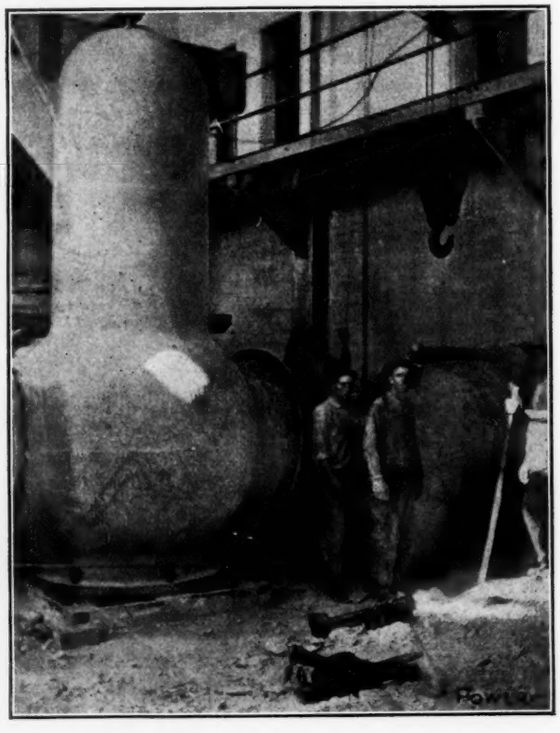


FIG. 1. SHOWING SIZE OF PIPE

would be as little moving as possible when making the final connections, to the pump.

An elevation of the suction well is shown in Fig. 2 with the floor above. In hoisting the pipe two tackles were used. The tackle block was inverted, and the hoisting rope passed through a snatch block to the hoisting crab. Two pieces of ⅝-in. steel cable were used for slings. Three turns were made around the pipe and two turns of the sling were passed over the hook of the tackle block, the third turn was made to haul taut to prevent slipping when lowering the pipe.

After lowering as far as conditions would allow, the load was shifted to the second tackle, and the hoisting tackle made fast, as shown by the dotted lines, Fig. 2. The elbow was then hoisted into position over the lower section of the pipe, a gasket put in place, and the pipes

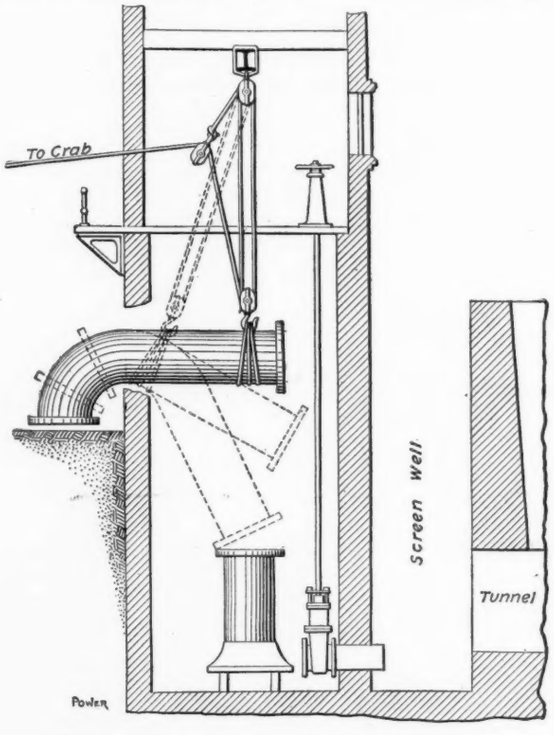


FIG. 2. PUTTING THE PIPE IN THE PIT

bolted together and connected to the condenser and stop valve. The suction pipe was moved to its correct position by hydraulic jacks. After the final connections were made, the opening in the wall was closed with cement.

JOHN F. NAGLE.

Troy, N. Y.

Problem in Power Plant Economics

A local company built an addition to its factory and at that time its plant consisted of a 5-ft. horizontal return-tubular boiler and a four-valve engine of 150 hp. This engine was running at about 90 r.p.m. and cutting off at ¼ stroke.

When the power equipment for the new addition came up for consideration, the management decided to use individual motors on the large machines and to group the small machines so as to be driven by one motor. A 250-kw. alternating-current generator was purchased and was to be driven by the same engine that was formerly used. Another 5-ft. horizontal tubular boiler was also purchased.

The heads of the firm were told at the time they bought this generator that the engine was working at its most economical point of cutoff and that it would not carry the load of this generator. The management claimed

that the engine would take care of the load from the new addition without any trouble.

As the new addition was started up one floor at a time the load came on gradually, but there finally came a day when the fireman could not hold the steam up to 110 lb. for more than a half hour at a time. The maker of the engine was then asked to send a man to indicate the engine.

The first diagrams taken showed that the engine was developing 230 hp. From the diagrams it was difficult to tell where the admission or cutoff took place, in fact, they could have been easily drawn with a square and a pencil.

The man from the engine manufacturer advised speeding the engine up to 118 r.p.m. and increasing the steam pressure to 130 lb. Now the question is, outside of danger from flywheel accident, due to overspeeding the engine, and knowing that the engine will do its work most economically at ¼ cutoff, is it cheaper to run it overloaded, taking steam at more than three-quarters stroke every revolution, or would it be cheaper to buy a new engine?

L. W. CHADWICK.

Bridgewater, Mass.

Combined Supply and Discharge for Tank

The illustration shows a simple arrangement of piping and fittings for a tank where it is desired to automatically prevent overflow, and at the same time permit the use of a single pipe to serve for both supply and discharge line.

Referring to Fig. 1, the pipe *A* extending through the bottom or side of the tank has on its inside end a balanced lever valve *B* operated by the float *C* in the usual manner.

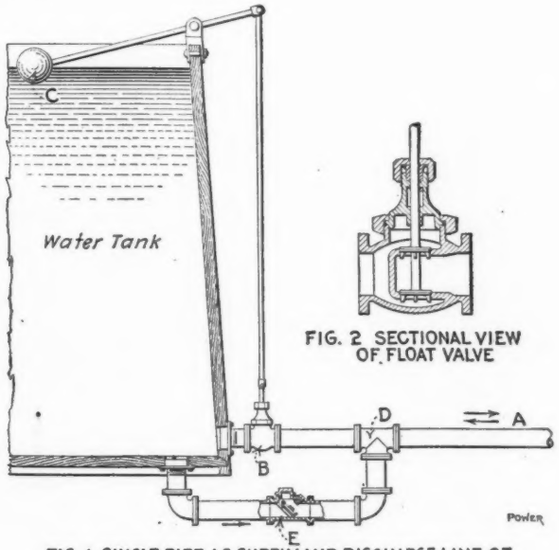


FIG. 1 SINGLE PIPE AS SUPPLY AND DISCHARGE LINE OF WATER TANK

FIG. 2 SECTIONAL VIEW OF FLOAT VALVE

Obviously if the float valve alone is used the pipe *A* could not be used as a discharge pipe after the float has closed the valve unless other means were used to open the valve. To overcome this difficulty the tee *D* is inserted and a bypass run into the tank, as shown. In the bypass there is a check valve *E* so placed that the water can flow from the tank, but not in the reverse direction.

When the water reaches its highest level in the tank both vavles will be closed, preventing overflow. If now the source of supply is cut off and an outlet from the pipeline opened the pressure will fall at the discharge end of the check valve, and the water will flow freely through the bypass as well as through the float valve when the float has dropped sufficiently to open it. A sectional view of the float valve is shown in Fig. 2.

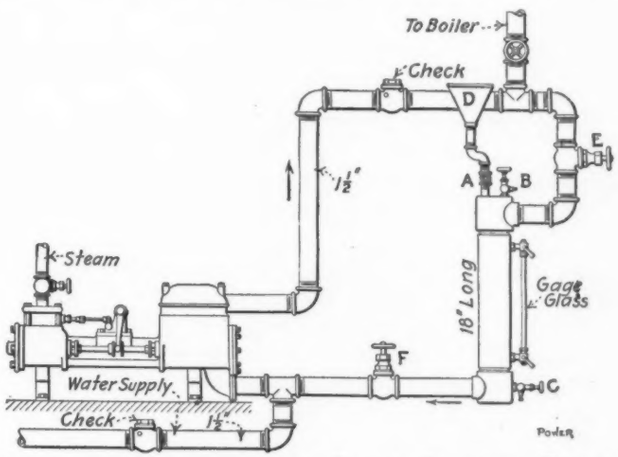
I have used this arrangement in an accumulator tank for gas-engine cooling water, providing for continuous operation while the circulating pumps are shut down for short periods, and in this case there was effected a saving corresponding to the cost of 700 ft. of 8-in. separate return pipe.

W. S. MAYERS.

Fairmont, W. Va.

Boiler Graphite Feeder

The accompanying illustration shows the receiver and piping made for use in connection with the boiler-feed pump to deliver graphite to the boilers. This arrangement works very well, as it not only feeds the graphite to the boilers in the most desirable manner, but it so well lubricates the packing in the pump that the packing gives from two to three times the service that it did before using graphite.



COMPLETE CONNECTIONS FOR BOILER GRAPHITE FEEDER

To fill the receiver, first open valve *A* and the air cock *B*, also open valve *C* to let the water out of the receiver. When it is empty, valve *C* should be closed. Mix the graphite with water until thin enough to pour through the funnel *D*. When the receiver is full, close the valves *A* and *B*, open valve *E*, and regulate the feed of graphite by opening valve *F* as much as is required.

As seen from the illustration, this 3-in. receiver and its connections may be easily made by the engineer from pipe and fittings found about the plant.

J. W. SPAULDING.

Seattle, Wash.

The American Society of Mechanical Engineers will underwrite $5000 or its proportionate share, in connection with the International Congress to be held in San Francisco in 1915, provided the other American societies make similar provision.

QUESTIONS BEFORE THE HOUSE

Four Pass Boilers

The plan of using four passes in steam boilers, referred to in the September 17 issue, is open to objections greater even than the greater draft required, the additional baffle to be kept tight, the larger setting and consequent greater infiltration of air, etc., which were mentioned.

Most important is that the additional surface will not as a rule pay for itself. This may be demonstrated from facts cited in the editorial, since it states that the additional fourth pass will reduce the temperature of the gases by 25 to 50° F. This may sound large in degrees but does not show up so much when reduced to percentage of fuel saved, for with the ratio of air to fuel ordinarily attained in good practice nowadays, 25 to 50° F. drop in the flue gases means only 1 to 2½ per cent. of fuel.

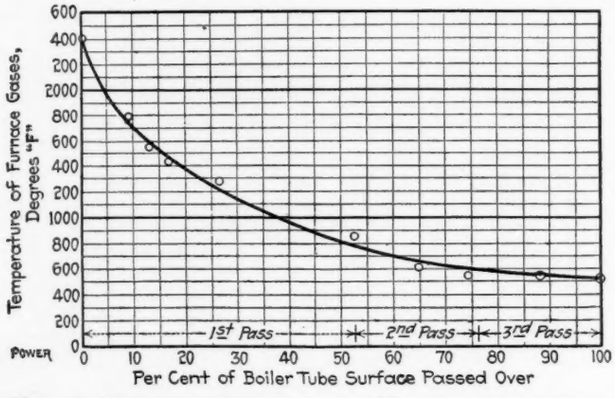


FIG. 1. CURVE OF TEMPERATURE VARIATION THROUGH A BOILER FURNACE

If the fourth pass be obtained simply by putting in more baffles in an existing boiler, that is reducing the surface in the first three passes to accommodate the fourth, it is doubtful whether the expected reduction in terminal temperatures will be realized, since there is no more boiler surface than before, and the only factor tending to a greater heat absorption is the increased velocity of the gases. On the other hand, greater draft will undoubtedly be required, since the intensity of draft required increases as the square of the velocity and as the amount of surface in the walls of the gas passages.

Fig. 1 shows temperature readings taken in a boiler of the B. & W. type, operated at about nominal capacity, that is, 10 sq.ft. to the boiler-horsepower, and fitted with three passes, the third and last pass containing some 23 per cent. of the total surface. If the thermometers are to be trusted the reduction in temperature in the third pass is only some 40°. I have figures relating to another boiler, and shown on the boiler outline Fig. 2, which, while not taken at so many different points in the path of the gases, agree fairly well with those given on the chart. As this second set of figures shows that the temperature at the end of the first pass was 800° F. and at the end of the third pass, 500°, about the same as in the first boiler; the distribution of the temperature between these two points probably coincided also. That a fourth pass would help in either boiler appears doubtful.

If the fourth pass is obtained by greater subdivision of the existing surface, the draft resistance will be increased considerably without a great increase in heat absorption, and, if we provide for the fourth pass by installing longer tubes and more surface per boiler horsepower, the question is: Will the extra surface repay its own upkeep charges? All information on the maintenance charges of boiler surface indicates that the answer should be "No".

Several authorities give 12 per cent. as the proper figure to be allowed for depreciation, repairs and upkeep on the boilers and equipment, and adding 5 per cent. for interest on the investment, the total charge comes to 17 per cent. The cost of boilers installed complete is rarely less than $2 per sq.ft., so that the annual charge on each square foot is 34c.

Assuming that the additional fourth pass is to reduce the flue gas temperature by 25 to 50° F., it will have to contain at least 2 sq.ft. of surface per boiler horsepower. (The low rate of absorption in the last pass, it might be mentioned, is due to the gases more nearly approaching the temperature of the water and steam within the boiler, so that the temperature head, to which the flow of heat from the gases to the water is directly proportional, is very low.) Two feet of surface at 34c. per ft. is 68c. per year annual charge. Granting that 2 per cent. of the coal will be recovered by these 2 ft. and assuming that 4 lb. of coal was originally

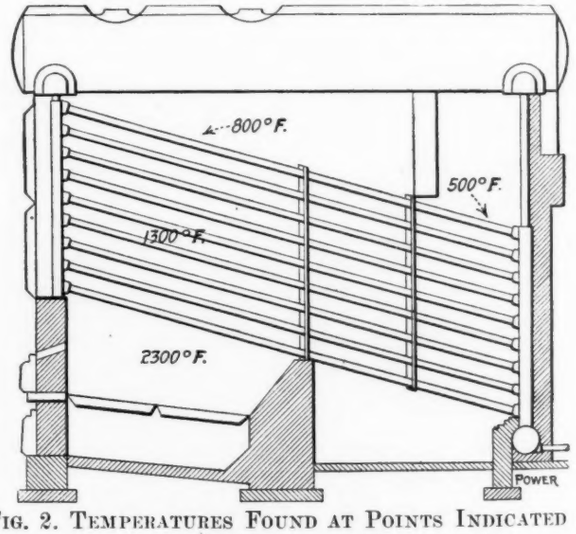


FIG. 2. TEMPERATURES FOUND AT POINTS INDICATED IN A WATER-TUBE BOILER

consumed per boiler horsepower-hour, and that the boiler is operated at full load for ten hours per day, 300 days per year, the coal saving, with $3 coal will amount to

$$\frac{4 \times 10 \times 300 \times 0.02 \times 3}{2000} = \$0.36$$

which is not enough to pay the annual charges upon the additional surface, not to mention the greater impediment to the draft, the greater difficulty of keeping the longer tubes tight and clean, the greater resistance offered to the circulation of the water through the tubes, etc.

The objection may be raised here that the deterioration of heating surface, in this last pass, subjected as it is to only moderate temperatures, will not be as great as in the rest of the boiler, but as the surface is in most boilers integral with those parts of the tubes exposed to the hottest gases, it will have to be discarded with them when they are burnt out and hence must bear the same charge for depreciation and repairs.

The editorial states that the increased draft through a four pass boiler will not, at the most, be greater than the draft necessary for a three-pass boiler with an economizer, which, taken with the remainder of the editorial, leads to the inference that the fourth pass is as good as, or to be preferred to, the economizer. Such, however, is far from correct, since an equivalent amount of economizer surface installed in place of the additional boiler surface would show much greater results, due to the lower temperature of its contents and consequently greater rate at which heat would be absorbed. If the temperature of the gases leaving the boiler is 500° F., and the temperature of the water and steam within the boiler is 370° F., a temperature head of only 130° is available. But pass the same gases through an economizer receiving water at 100° F., and a maximum temperature head of 400° is possible. In other words, per square foot, the economizer surface will recover three times as much heat, and since it costs less in the first place and does not deteriorate so rapidly, will pay more than three times the return. As a matter of fact, in installing an economizer it would not be good practice to stop with two square feet per boiler horsepower since satisfactory profits can be earned upon a much larger amount of surface. In the present example it would probably pay to reduce the temperature of the gases to 300° F., raising the temperature of the feed water to 200° F.

Furthermore, the most recent practice appears to be toward curtailing boiler surface, rather than increasing it, and many designers believe that it does not pay to install more than 4 to 6 sq. ft. per boiler horsepower output at maximum load. Under these conditions the gases will leave the boiler at 575 to 650° F. and the feed water will be delivered to the boiler from the economizer at upward of 250° F.

New York City. GEO. H. GIBSON.

Importance of Furnace Efficiency

In the article by Joseph Harrington, which appeared in the Dec. 17 issue of POWER, the argument was made that the heat balance from a boiler trial should be made on the basis of coal as fired. While computations based on coal as fired, including the moisture in the fuel would give the true efficiency of the boiler, furnace and grate for one particular case, the results could not be applied to compare the economy of a boiler with the same fuel but with different equipment; neither could the fuel under test be compared with other fuels. The amount of moisture in a fuel as fired depends to a considerable extent on conditions of transportation and storage which cannot be controlled. For the correct comparison of boiler trials the various economy and efficiency results should be based on the dry fuel as fired. Then, to represent the results under the conditions of the test as actually carried on, the economy and efficiency computations should also include results on coal as actually fired, including moisture.

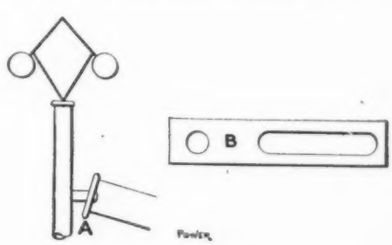
The boiler code of 1912 submitted by the Power Test Committee of the American Society of Mechanical Engineers gives the economy results in terms of the evaporation from and at 212 deg. F. per pound of wet coal fired, per pound of dry coal and per pound of combustible. These same comparisons should be carried out under the heading of efficiency and also the final heat balance should be based on wet coal as fired, on dry coal and on the combustible. The combustible is obtained by subtracting the ashes and other noncombustible matter in the coal, which passes through the grate or is removed during cleaning, from the dry coal.

A. A. POTTER.

Manhattan, Kan.

Equalizing Cutoff without Dashpot Troubles

In the Nov. 5 issue, E. P. Haines discusses the matter of equalizing the cutoff throughout the range of the governor. He tells how this was accomplished, but states that the trouble with the dashpots was not entirely overcome even with the special construction of the knockoff-cam-arms and the head-end hook lever. If Mr. Haines will turn to page 293, of the Feb. 9, 1909, issue of POWER,



he will find a very simple means for getting an equalization of cutoff without any annoyance whatever from dashpot trouble.

G. D. LIVINGSTON.

Minneapolis, Minn.

[The manner of equalizing the cutoff, referred to, as described in the letter in the Feb. 9, 1909, issue, is, essentially, as follos:

To get the crank-end valve to stay open longer at full load and close sooner at light load, the lower end A of the governor rocker-arm, operating the cam of the crank-end valve was made longer. To determine the proper length of the lower end of A, a piece of iron B, about ⅛x⅝x3 in., was drilled to be bolted to the arm. The slot was used to attach the governor reach-rod. By taking a diagram at a light load and one at a heavy load the necessary length was easily determined. A new rocker-arm was then cast, the lower end of which was made as long as the experiment indicated was necessary.—Editor].

INQUIRIES OF GENERAL INTEREST

Boiler Walls—Should the walls of a boiler setting be hollow or solid?

W. E. T.

Recent tests by the Bureau of Mines indicate that there is no advantage in hollow over solid walls. The latter show less loss of furnace heat by convection than hollow walls.

Valve Slam—How can "valve slam" on the water end of a pump be prevented?

H. W. W.

By putting an air chamber on the suction pipe. If the intake water is received under pressure, the chamber should be put near the pump. If the water is taken from a well or cistern, a foot valve should be used on the suction pipe and the air chamber connected near the surface of the water.

Stopping Corliss Engines—What is the proper way to stop and start a high-speed Corliss engine? Should both the steam and exhaust-valve rods be unlatched?

G. F.

To stop a single Corliss engine, so that it will be off center when it comes to rest, it is advisable to unhook the wrist plate. If it is a double eccentric engine, it is usually only necessary to unhook the steam valve when stopping or starting.

Advantage of Smoke Flue Covering—What would be the atvantage of covering the flue and uptake of a boiler with 2 in. of asbestos?

F. G.

It would make the boiler room cooler and might increase the capacity of the stack. The best draft is with a chimney temperature of 620 deg. F., and, if the gases are now delivered to the stack at less than that temperature, the velocity of the chimney gases would be increased. If, as is more than likely, the stack temperature is already above 620 deg. F., there would be no advantage in the covering other than to make the boiler room cooler.

Pump Lifting Warm Water—If a plunger pump is placed close to its supply tank and 6 ft. above it, can it lift water at 150 deg. F.?

J. P. F.

Neither a plunger pump nor any other type can "lift" water under pressure of the atmosphere at 150 deg. F., a height of 6 ft., because the vapor given off by the water at that temperature when the pressure above it is reduced, destroys the vacuum created by the pump. Sufficient difference between the pressure of the atmosphere acting on the surface of the water in the tank and that in the suction pipe cannot be maintained to raise the water 6 ft.

Pump Horsepower—What horsepower is required to pump 210 gal. of water per minute against a pressure of 86.8 lb.?

N. K. F.

The water pumped per minute, 210 gal., is equal to

$$210 \times 231 = 48{,}510 \text{ cu.in.}$$

Pumping 12 cu.in. against 86.8 lb. per sq.in. pressure is doing 86.8 ft.-lb. of work. Therefore, pumping 48,510 cu.in. per min. against this pressure is a work of

$$\frac{48{,}510 \times 86.8}{12} = 350{,}889 \text{ ft.-lb. per min.,}$$

or,

$$\frac{350{,}889}{33{,}000} = 10.6 \text{ hp.}$$

Running Lap-Jointed Belts—When a belt has a cemented lap joint, which way should the lap point with respect to its direction of motion?

H. N. Q.

Many prefer to put the end of the inside lap so that it travels toward the pulley, then any slip of the driving pulley would tend to wipe the joint down to position. However, if the driven pulley is the one inclined to slip, the lap should be run the opposite way. The same is true if the belt is where it is likely to be wet and the point of the lap become loose from the inside of the belt, as then it would be apt to be turned up or away from the belt as it strikes either pulley. Generally, the governing condition is which pulley is most inclined to slip and then the lap should be run in the direction that will tend to keep it laid against the belt.

Burning Wood and Coal—With which is there the least waste of combustible matter, in the burning of wood or coal?

J. D.

Ordinarily, there will be less waste of combustible matter when burning wood under a boiler, for, when burning coal, there is more chance that some of the combustible matter will not be reached by the heat of the fire. Recovery of the good fuel passing through the grates is easier with wood than with coal, as the unburned part is more easily separated from the ashes. With a proper regulation of the air supply, there can be as complete combustion with one fuel as with the other.

Safety Valve Size—What governs the proper size of a safety valve for a given boiler?

N. A.

This is determined by the grate area and the pressure to be carried on the boiler. A very good formula for safety valve size is:

$$A = \frac{37.5 \times G}{P}$$

where

A = Area of valve in sq.in.;
G = Area of grate in sq.ft.;
P = Absolute pressure in lb. per sq.in., or gage pressure plus 15 lb.

Compression Space in Pump—What amount of cushion space is required in a pump to prevent the piston striking at the end of the stroke?

W. H. W.

The proper cushion space will depend on the speed of the pump and the weight of the reciprocating parts. No rule can be laid down for calculating the compression space. It must be determined for every design and service of pump by experience. In some pumps, such as fire pumps, which are driven at a high piston speed, the compression can be regulated by adjusting the compression or cushion valves to permit more or less of the compressed steam to escape to the exhaust. The first item on page 672 of May 7, 1912, issue, illustrates the cushion valves of a pump cylinder and explains their function and action.

Testing Squareness of Engine Shaft—How can the shaft of a horizontal engine be tested to see that is square with the guides and the center line of the cylinder?

J. R. C.

Stretch a horizontal line (fish line is best) taut over the center of the guides and cylinder, making it fast to fixed supports at each end. Test it for exact position by dropping a plumb line beside it (always on the same side) and shift it until it is exactly parallel with the engine center line. Then stretch another horizontal line at right angles with the first and touching it, putting it as nearly as possible with the eye directly over the center of the shaft. To get the two lines perpendicular to one another, measure from their intersection 3 ft. along one line and 4 ft. along the other. Mark these two points by sticking needles through the lines. When the lines are at right angles, the distance between the needles will be exactly 5 ft. Any other multiples of these dimensions, 3, 4 and 5, as 30, 40 and 50, may be used, or any distances bearing the same ratio. When the two lines are at right angles, a plumb line dropped from the line over the shaft touching it (always on the same side) should hang the same distance from the center line of the shaft wherever it is dropped. If it does not, the outboard bearing should be shifted until it does.

ENGINEERS' STUDY COURSE

MENSURATION—PART IV

POLYGONS

A *polygon* is a *plane figure* inclosed by any number of straight lines.

Thus triangles, rectangles, hexagons and octagons are examples of polygons. A polygon is said to be *regular* when all its sides and the angles formed by the sides are equal. A regular polygon is shown in Fig. 1. An *irregular* polygon is one whose sides or angles are not all

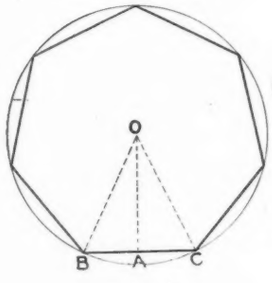


FIG. 1

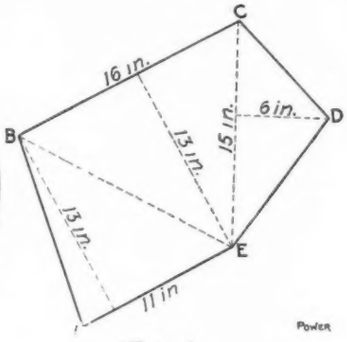


FIG. 2

equal. Such a polygon is shown in Fig. 2.

The *perimeter* of a figure is the total length of its sides or boundary.

At a glance we can see that it is much easier to measure the area of Fig. 1 than of Fig. 2, because all the angles in Fig. 1 are equal, and, having laid off the triangle *BCO* and found its area by multiplying *BAC* by one-half of *AO*, we simply multiply by the total number of triangles to get the area of the polygon. With the

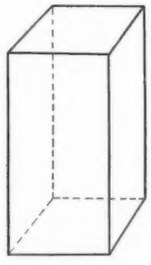


FIG. 3

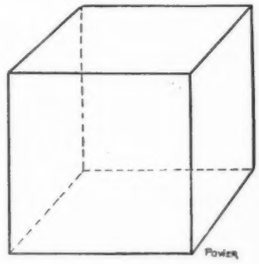


FIG. 4

irregular polygon, we must divide it into triangles, as was done in Fig. 2, before we can readily find the area. The student should always study closely an irregular polygon before laying off the triangles, as the aim is to make as few triangles as possible.

Let Fig. 2 represent a piece of sheet iron and let it be required to find the area.

We have three triangles, with the areas as follows:

Area *BAE* is

$$\frac{11 \times 13}{2} = \frac{143}{2} = 71.5 \; sq.in.$$

Area *EBC* is

$$\frac{16 \times 13}{2} = \frac{208}{2} = 104 \; sq.in.$$

Area *DCE* is

$$\frac{15 \times 6}{2} = \frac{90}{2} = 45 \; sq.in.$$

The total area of the polygon is

$$71.5 + 104 + 45 = 220.5 \; sq.in.$$

This method may be applied to any irregular polygon.

Suggestion: Find what it would cost to lay a concrete floor for an engine room, shaped as an irregular polygon. The cost of such flooring will average 25c. per sq.ft.

PRISMS AND CYLINDERS

A *prism* is a solid, whose bases are polygons and whose sides are parallelograms.

Prisms are of many kinds, i.e., they are rectangular, triangular, hexagonal, etc., according to the form of their

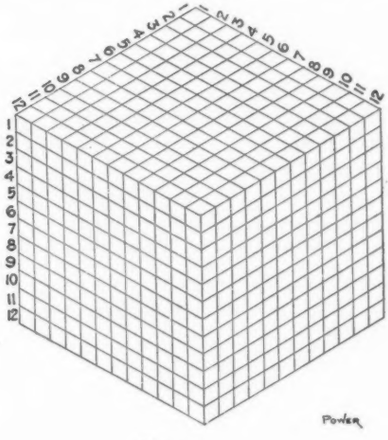


FIG. 5

bases. Many water and other kinds of tanks may be considered as prisms. Rectangular prisms are shown in Figs. 3 and 4. Fig. 4 is called a cube.

A *cylinder* is a body whose ends form parallel circles and which is of uniform diameter. A cylinder is not a prism.

Areas are figured in square measure, as square inches or square feet and volumes are figured in cubic measure, as cubic inches or cubic feet.

We learned in our lesson on denominate numbers, Nov. 26 issue, that a unit of volume such as the cubic inch, is the volume of a cube whose sides are 1 sq.in.

To illustrate this unit of volume, consider the cube, Fig. 5, the edges of which measure 12 in., the area of a side will be

$$Area = base \times altitude$$

or

$$12 \times 12 = 144 \; sq.in.$$

We see that the cube is divided into twelve slices, each slice containing 144 sq.in. As Fig. 5 has twelve such

slices, containing 144 sq.in. each, the whole cube, or cubic foot, contains

$$144 \times 12 = 1728 \ cu.in.$$

Thus, to reduce cubic inches to cubic feet, we divide by 1728 and to reduce cubic feet to cubic inches, we multiply by 1728.

From the above, we may form the rule for finding the volume of a prism or of rectangular vessels. The term *cubical contents* means the same as volume.

Rule: *Multiply the area of the base by the altitude.*

Example: How many gallons of water will a rectangular water tank 18 ft. long, 7 ft. high and 10 ft. wide contain? One U. S. gallon contains 231 cu.in.

The volume of the tank is

$$18 \times 10 \times 7 = 1260 \ cu.ft.$$

The number of gallons contained is

$$\frac{1260 \times 1728}{231} = 9425.45 \ gal.$$

Example: If there is a load of 1236 lb. on the supports holding the empty tank, what will be the load supported when the tank is full of water? Water weighs 8.33 lb. per gal. at 62 deg. F. The total load is

$$(9425.45 \times 8.33) + 1236 = 79{,}750 \ lb.$$

The *lateral area* of a prism is the total area of its sides or faces, except its ends or bases.

To find the lateral area.

Rule: *Multiply the perimeter of the base by the altitude.*

Example: We are to line the tank in the previous example, with zinc. Allowing 22 sq.ft. additional area for the overlap at the seams, how many square feet of zinc will we need?

The perimeter being

$$(18 + 10) \times 2 = 56 \ ft.$$

the lateral area is

$$56 \times 7 = 392 \ sq.ft.$$

Adding the area of the bottom

$$18 \times 10 = 180 \ sq.ft.$$

and the 22 sq.ft. for the seam laps, the total area of zinc needed is

$$392 + 180 + 22 = 594 \ sq.ft.$$

The *convex area* of a cylinder is equivalent to the area of the rectangle that would be formed by spreading out the side of the cylinder, as in Fig. 6, although in Fig. 6 only one-half of the side is shown spread out. We see,

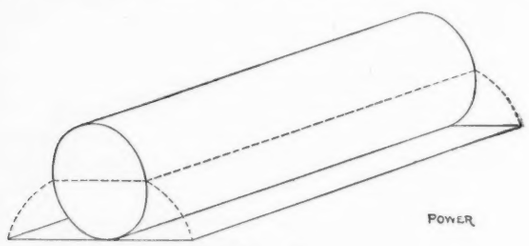


FIG. 6

from this, that the rule for finding the convex area of a cylinder will be

Rule: *Multiply the circumference by the altitude.*

Example: For surface condensers serving turbines, 3.5 sq.ft. of cooling surface is sometimes allowed per kilowatt. According to this rating, what capacity turbo-generator could be served by a surface condenser having 7020 tubes 14 ft. by 0.75 in.?

The circumference of one tube is

$$0.75 \times 3.1416 = 2.36 \ in.$$

The convex area of one tube is

$$14 \times 12 \times 2.36 = 396.48 \ sq.in., \text{ or } 2.75 + sq.ft.$$

The capacity of the condenser would be

$$\frac{7020 \times 2.75}{3.5} = 5515.6 \ kw.$$

Example: At a steam consumption of 12.8 lb. per kw.-hr. for the turbine, how many pounds of steam would the condenser handle per hour?

$$5515.6 \times 12.8 = 70{,}609.68 \ lb. \quad \text{Ans.}$$

EXAMPLES FOR PRACTICE

(1) (a) How many gallons of water will be contained in a water-works standpipe 18 ft. in diameter by 125 ft. high?

(b) Making no allowances for losses, in what time will the standpipe be filled by a duplex pump having 30-in. plungers, 34-in. stroke, each of the two plungers making 18 strokes per minute?

A rule sometimes given for finding the water capacity of a return-tubular boiler is:

Find ⅔ the volume of the shell and from this subtract the volume of all the tubes.

(2) Example: What is the water capacity in cubic inches of a boiler 18 ft. by 72 in. with 72 three-inch tubes?

(a) Capacity in gallons?

(b) Capacity in pounds?

Due to an error in pointing off the number of decimal places in the example (Feb. 4 issue) for finding the area of a safety valve, by the Massachusetts formula, an answer of 2½ in. instead of 3.77 in. was obtained for the diameter of the valve.

As applicants for engineers' licenses are usually called upon to work out this formula, the example given in the Feb. 4 issue is fully worked out below.

Massachusetts safety-valve area formula:

$$A = \frac{W \times 70 \times 11}{P}$$

A = Area of valve in square inches for each square foot of grate surface;

W = Pounds of water evaporated per second per square foot of grate surface;

P = Absolute pressure per square inches at which valve is to open.

Example: A boiler has 30 sq.ft. of grate surface, a combustion rate of 20 lb. of coal per square foot of grate per hour, evaporation of 10 lb. water per pound of coal; gage pressure 100 lb. What size safety valve is required?

$$30 \times 20 = 600 \ lb.$$

coal burned per hour.

$$600 \times 10 = 6000 \ lb.$$

water evaporated per hour.

$$\frac{6000}{3600} = 1.67 \ lb.$$

water evaporated per second.

$$1.67 \div 30 = 0.05567 \ lb.$$

water evaporated per square foot of grate per second.

Substituting, we have,

$$A = \frac{0.05567 \times 70 \times 11}{115} = 0.3728 \ sq.in.$$

area of valve for one square foot of grate.

As there are 30 sq.ft. of grate, the area of the valve will be:

$$0.3728 \times 30 = 11.184 \text{ sq.in.}$$

The diameter of a valve of 11.184 sq.in. area, will be

$$\sqrt{\frac{11.184}{0.7854}} = 3.77 \text{ in.}$$

A 4-in. valve would be the proper size valve to install.

❖

Boiler Capacity Required to Heat a Swimming Pool

By J. P. Lisk

As a general rule the operating engineer of a power plant is not interested in the design of swimming pools, but he is interested in the boiler power it requires to heat one up ready for use, if it so happens that he has such a heat consumer attached to his plant. Heating the water in the pool from 40 to 80 deg. F. is the average requirement in practice, and the time taken varies from four to eight hours, depending on local conditions.

A boiler horsepower is equivalent to the evaporation of 34.5 lb. of water from a feed-water temperature of 212 deg. F. into dry steam of the same temperature. A British thermal unit is the quantity of heat required to raise the temperature of one pound of water one degree at its greatest density. These units were used in determining the constants in the following rules for finding the boiler horsepower required in any case in which the rise in temperature is 40 deg.

When the quantity of water in the pool is given in pounds, multiply by 0.0012.

When in gallons, multiply by 0.01.

When in cubic feet, multiply by 0.075.

In each case divide by the number of hours in which the work is to be done.

When the rise in temperature is more or less than 40 deg., the formulas given below may be used.

$$Hp. = \frac{(t_2 - t) \times h \times P}{U \times T} \quad (1)$$

$$Hp. = \frac{(t_2 - t) \times h \times a \times G}{U \times T} \quad (2)$$

$$Hp. = \frac{(t_2 - t) \times h \times b \times C}{U \times T} \quad (3)$$

t = Initial temperature of water;
t_2 = Final temperature of water;
h = Quantity of heat to raise one pound of water one degree in temperature = 1 B.t.u.;
U = Units of heat in one boiler horsepower = 33,479;
$Hp.$ = Boiler horsepower to do the required work;
T = Time in hours in which the work is to be done;
P = Weight of water in pounds;
G = Quantity of water in gallons;
C = Quantity of water in cubic feet;
a = Pounds in gallon of water = 8.335;
b = Pounds in cubic foot of water = 62.35.

Formula (1) is to be used when the quantity of water is given in pounds, formula (2) when in gallons and formula (3) when the quantity is expressed in cubic feet.

OVER THE SPILLWAY

JUST JESTS, JABS, JOSHES AND JUMBLES

Out on the Pacific Coast the engineering folks are much perturbed over what they call the "Hetch Hetchy" project. Just what is this project, and why the perturbation? We've heard of the hootchee-kootchee, read of the hari-kari and fraternized with the hunyadi, but somebody must put us hep to this hetch hetchy thing. Anything like beri-beri or tummyache?

❖

Apropos of ailments, the Civic Federation wise guys say we of the laboring classes are physically going into the discard if we don't watch out. There's boiler maker's deafness, hatter's shakes, potter's rot, painter's colic and wrist-drop, caisson worker's bends, match maker's phossy-jaw, brass worker's chills, glass blower's cataract and miner's asthma. Phew! We're some ailing. Skipping the hookworm, who overlooked the blind staggers, the botts and the housemaid's knee?

❖

Sydney Pickles (real name) has pulled the heinz over all the other well known varieties by "reaching a height of 7200 ft. in his airship." No, Eustace, there was nothing the matter with Pickles' feet as he was walking down the street. When you saw Syd, he was bound for the hangar—you know, the stable where they tie up an airship and bed it down.

❖

Mr. Livergood has resigned as chief engineer for the Central Illinois Public Service Co.—"Electric Railway Journal." Now, don't get personal. We are just turning the lights on Mr. Livergood, as he is going into the electrical business for himself. May he live a good long life, and prosper.

❖

This is "old home" week with our engineering humorists, and proves that your Uncle Spill is not all there is to a jingling jest and joyous jumble. We have with us this evening, Brother Pattison, of Lexington, Ohio, who will say a few words:

Hex Nut—How's the old head, Cy Linder? Where were you yesterday?

Cy—I went to the races. I saw the Engine race and the Mill race, but best of all was the Run of the Mine. Anthracite Coal was coming up hard when she leaped out of the shafts and landed in a heap on the track. They took An out to crush'er and she's been nutty ever since. An's brother Bi was also entered, but he was too soft—just in from the fields. A peculiar thing about Bi is that when given the rain he slacks up. Bi got away with the big lump, though, and you ought to have heard them roast him in the Coke Oven event. Bi thinks that smoking should not be allowed in a furnace, bue he stacked up pretty well there except when the Boiler flew. After the final heat, everybody said that Bi was the best that ever hit the dump.

❖

Here's one, I see, from Tennessee: Lloyd V. Beets raised it in his garden and sends it to market:

A certain salesman was noted for his habit of occasionally taking a week "off" when the Tennessee mountain dew got within his midst.

Finally, his boss lost patience and wrote: "I've preached, sworn and prayed that you would take an interest in this firm. What do you intend to do?"

The quick-witted and dew-wetted salesman sent this telegram: "I'll take an interest. Don't change name of firm until I return.

❖

Here's a tale of a soft corn that is hard to believe, from Framingham, Mass.:

Taddy, the long, lean assistant engineer at the power house, had a corn on his toe so tender that when on duty he had a shoe and a stocking especially cut to keep down its temperature within comfortable bounds. He was just as sensitive as the corn was about it. The other day he was putting in a new carbon on the live side of a circuit-breaker. He stood on a chair, and the corn protruded over the edge in dangerous proximity to the main switch. I was about to warn him when he moved. There was a flash and a yell, and Taddy lay stunned on the floor. Now, he wears a whole shoe and that particular corn bothers him no more. Wonderful, what electricity accomplishes these days, ain't it?

Large High Vacuum Condenser Test

By J. J. Brown

SYNOPSIS—Data pertaining to a test of a large high vacuum condenser which was guaranteed to produce as high as 28.9 in. of vacuum with 70 deg. cooling water. The test shows that this guarantee was exceeded. Charts and tables and details of the test are given.

The test figures and curves given herewith, demonstrate the success that has been achieved in solving the problem of high-vacuum condenser design.

Vacuums from 28.3 to 28.9 in. were guaranteed with 70 deg. inlet cooling water. The small margin between these vacuums and the maximum attainable can hardly be realized without citing a few figures from the steam table. If the cooling water were to be heated only one degree in the condenser (that is, from 70 to 71 deg.), then the maximum vacuum obtainable would be 29.23 in. If it were heated to 80 deg., the maximum vacuum obtainable would be 28.97 in.; if heated to 90 deg. 28.58-in. the vacuum would be the maximum.

It is evident, therefore, that the task imposed upon the condenser designer by the demands for high vacuum with steam turbines, has been a severe one. The surface must transmit heat at very high efficiency because the available temperature difference or "heat head" causing the flow of heat is but a few degrees. The air and noncondensible gases are at an even lower pressure than that of the turbine exhaust and the greatest refinement must be observed in the manufacture of the air pump, in order that it may successfully withdraw these highly rarified gases from the condenser.

There are many other very interesting phases of the problem of high-vacuum condenser design, as, for instance, the proper arrangement of dry plates to prevent flooding of the tube surface with condensate or "steam water" and to maintain high temperature of condensate; the arrangement of tube surface to give high velocity of water through the tubes; the design of the circulating pump and circulating system so as to cause the least consumption of power in pumping the cooling water; and finally, the arrangement of tube surface to give efficient steam distribution.

At high vacuums, the specific volume of steam becomes enormous and the addition of tube surface to a condenser of ordinary design, may entirely defeat the purpose of the designer because of the increased resistance to the flow of steam between the tubes.

The loss of head through an orifice varies as the square of the velocity and as the first power of the density. For a constant weight of steam per second, corresponding to a constant turbine load, the resistance varies directly as the square of the volume and inversely as the first power of the volume. In other words, the resistance varies directly as the first power of the volume. The increase in volume with increase in vacuum, is indicated by the following figures:

Vacuum	Specific Volume*
26	176.7
27	231.9
28	339.6
29	657.0
29.5	1259.0

*"Steam Tables for Condenser Work," published by Wheeler Condenser & Engineering Co.

At a 29-in. vacuum the resistance flow is almost four times the resistance at 26 in.

Furthermore, an increase in vacuum means lower rate of condensation per square foot of surface. Accordingly with an increase in vacuum there is less condensation on the first rows, and a greater weight of steam must flow through the restricted area between the tubes at the top of the condenser. This additional flow causes increased resistance and back pressure, or lower vacuum.

An attempt to obtain better vacuum by the addition of tubes may thus be entirely unsuccessful, unless provision has been made for efficient distribution of the steam to all of the tube surface without undue resistance and back pressure.

These considerations lead to a very flat condenser of small height, covering a very large area. But such a design requires too much floor space and has an added drawback in that the noncondensible gases are spread throughout the lower region of the condenser in a sheet of large area, whereas for efficient removal, the air should be concentrated at a few points. The advantages of the large, flat condenser without its disadvantages, are secured by the use of the rectangular shell and the proper arrangement of tubes and baffles.

Tubes from various portions of the top of the condenser are omitted, thus providing passageways to conduct the steam to a number of short banks of tubes and insuring its penetration to all of the surface. Distribution of the steam to the surface in this manner also prevents dead air spots, since the various banks of tubes are short and the circulation is vigorous and positive. The air is swept on to the lower part of the condenser where it passes over tubes arranged to cause brisk circulation of the steam-air mixture, so that when it enters the air-pump suction, the steam vapor has been removed to a large extent, and the mixture is rich in air. To further increase the density of the air, and to remove the water vapor, independent air coolers are sometimes installed, especially in large installations of the kind to be described.

Condensing Equipment Tested

This condenser is of the dry-tube base design, manufactured by the Wheeler Condenser & Engineering Co., Carteret, N. J., and is installed at the south works of the Illinois Steel Co., South Chicago, Ill., in connection with a 7000-kw. Curtis mixed-pressure turbo-generator. Fig. 1 shows the arrangement of turbine, base condenser and condenser auxiliaries in plan and elevation. Fig. 2 shows the turbo-generator, Fig. 3 the centrifugal circulating pump and Fig. 4 the rotative dry-vacuum pump.

Low-pressure steam goes to the turbine from three 2000-kw. compound, reciprocating engines, which discharge into a header about 100 ft. long. The steam enters the turbine at a few pounds above atmosphere, having passed through a large separator (shown behind the air pump in Fig. 4), which is designed to remove all but a small percentage of the moisture. In addition to this supply of steam, the exhaust from blast-furnace blowing engines and auxiliaries is used up to a total turbine

capacity of 200,000 to 250,000 lb. per hour. High-pressure steam to carry the load in case of a deficiency in the supply of exhaust steam, is obtained from the boiler plant, the piping being seen above and to the right of the turbo-generator in Fig. 2.

The turbine rests upon the condenser as a base. The tube surface is arranged in two sets of banks on either side of the center line with dry plates at several elevations on each side of the center which drain outward toward the shell. They divide the tube surface into a number of compartments, designed to give the most efficient and even distribution of the steam to all of the surface.

There are about 6000 tubes of 1-in. diameter, making in all about 25,000 sq.ft. of surface. The circulating water makes two passes through the tubes, entering at one end and passing through the lower bank of tubes on both sides of the center, returning through the top bank to the discharge.

The circulating system is closed, the water being drawn from Lake Michigan through an intake tunnel 150 ft. long and discharged to a sewer. The head on the circulating pump is from 10 to 20 ft., and the quantity of water pumped is from 24,000 to 34,000 gal. per min. Of this head, about half is lost in the condenser. (See test curves in Fig. 5.) The circulating pump is a Wheeler 36-in. double-suction volute pump, operating at a speed of from 100 to 135 r.p.m. driven by a 20x24-in. Cooper Corliss engine, seen in the background of Fig. 3.

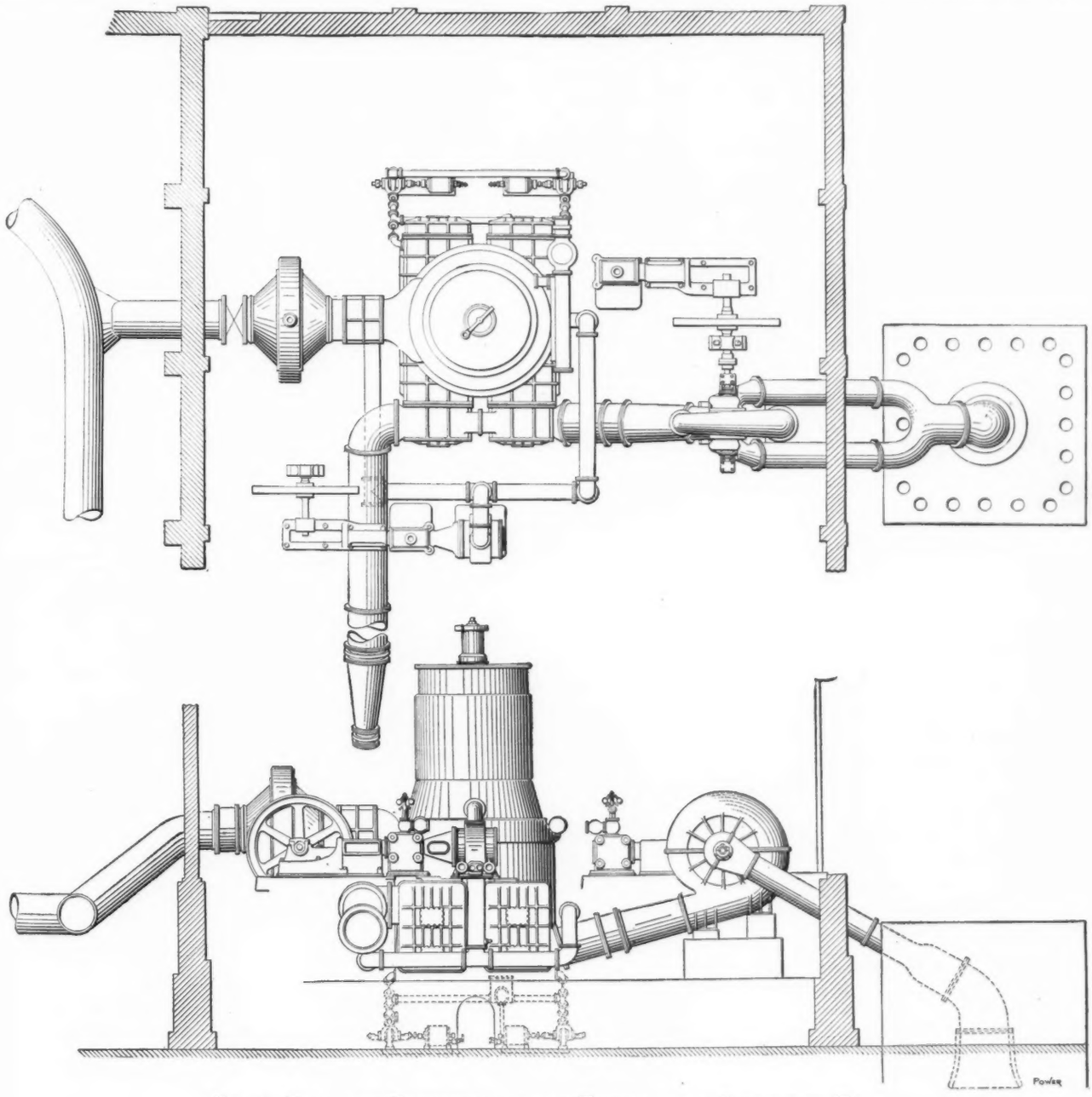


FIG. 1. PLAN AND ELEVATION OF THE TURBINE AND CONDENSER UNIT

The 14 and 36 by 24-in. Wheeler tandem rotative, dry-air vacuum pump has shown remarkable efficiency, as indicated by the tests of Table 1. The air is withdrawn near the bottom of each side of the base condenser by two air pipes running horizontally and joining opposite the

end of the condenser where they meet a pipe which rises through the engine floor, making a vertical loop before connecting to the pump. Within the vertical riser is a specially designed air cooler.

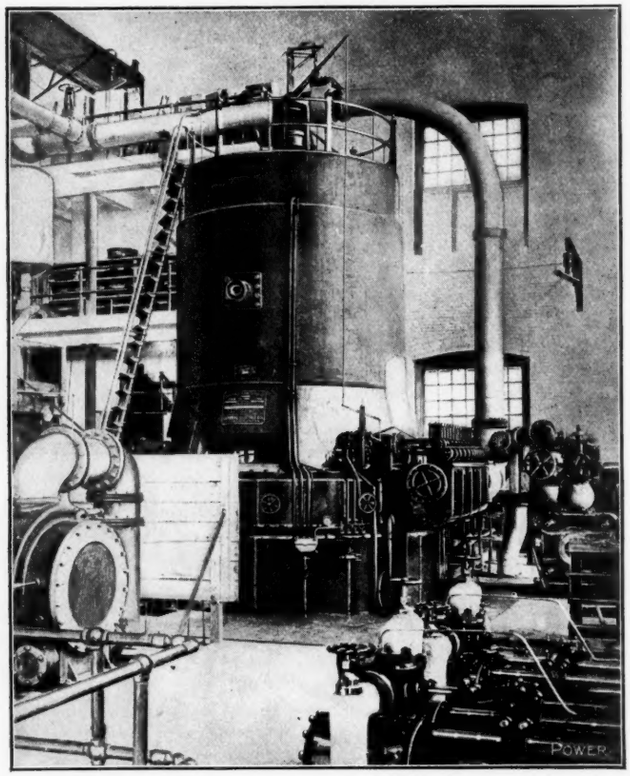


FIG. 2. 7000-KW. CURTIS TURBO-GENERATOR

Two 5-in. centrifugal hotwell pumps are installed, driven by 24-hp. steam turbins.

GUARANTEED PERFORMANCE AND RESULTS OF TEST

The guarantees made on the condensing equipment were as follows:

Temperature of cooling water 70 deg. F.

Quantity of water, not more than 34,000 gal. per min.

Vacuum	Load in lbs. of Steam per Hour
28.3	210,000
28.5	185,000
28.6	150,000
28.7	120,000
28.9	70,000

A summary of the tests made by the steam-engineering department of the Illinois Steel Co., to check the guaranteed values, is given by Table 2. The tests were made in July, 1910, with inlet circulating water at 73.4 to 73.7 deg. F. In calculating the final results, the performance was corrected to 70 deg. water, and a smooth curve drawn between vacuum and load is given in the chart of Fig. 6. The same chart also gives the actual vacuums with the water averaging about 73.6 deg. F.

TABLE No. I. TEST OF WHEELER R.D.V. PUMP. ILLINOIS STEEL COMPANY—JULY, 1910.

		Steam Cylinder		Air Cylinder		
Time	Speed	M.E.P.	I.H.P.	M.E.P.	I.H.P.	Mech. Eff.
3:21	81	32.8	61.9	5.40	53.7	86.8
3:26	93	35.0	75.6	5.55	63.3	83.5
3:30	93	34.15	74.0	5.25	60.0	81.1
3:35	94	32.5	71.1	5.37	62.0	87.1
3:40	93	34.0	73.6	5.43	62.0	84.2
3:45	94	33.7	73.8	5.36	61.8	83.7
3:50	94	33.5	73.3	5.48	63.2	86.2
3:55	93	34.25	74.2	5.46	62.4	84.0
4:00	93	34.5	74.7	5.33	60.9	81.5
4:05	94	33.25	72.8	5.55	64.0	88.0
4:10	93	34.0	73.8	5.36	61.2	83.0
4:15	94	35.0	76.6	5.36	61.8	80.7
4:20	94	34.5	75.5	5.43	62.6	83.0
4:25	93	34.0	73.6	5.36	61.2	83.2
4:30	94	34.1	74.6	5.31	61.2	82.0
4:35	93	33.35	72.2	5.25	60.0	83.1
4:40	93	33.35	72.2	5.37	61.3	84.9
4:45	93	34.4	74.3	5.33	60.8	81.7
4:50	94	34.35	73.0	5.40	62.3	85.3
4:55	93.5	32.5	70.6	5.36	61.5	86.9
5:00	94	33.45	73.2	5.28	60.9	83.2
5:05	94	34.1	74.6	5.22	60.2	80.7
5:10	94	32.7	71.6	5.19	59.8	83.5
Average	93.0	33.76	73.1	5.37	61.2	83.89

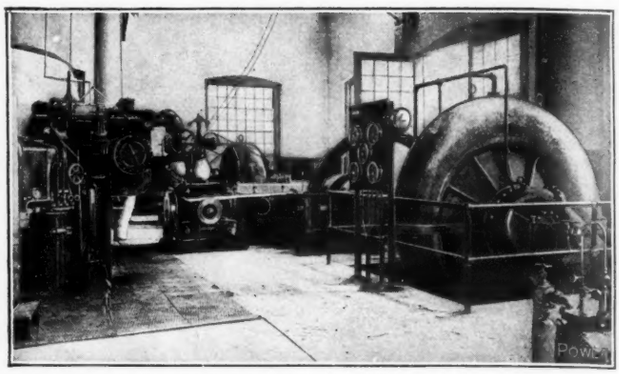


FIG. 3. CENTRIFUGAL OSCILLATING PUMPS

According to readings taken at the time of the condenser tests, the quantity of water was slightly under 35,000 gal. per min., and in calculating the vacuum, correction was made to 34,000 gal. per min. capacity, as the guarantees were made upon this basis. It should be stated here, however, that subsequent investigation led to the discovery of an error in the meter for measuring the circulating water, so that the actual quantity pumped was somewhat less than 34,000 gal. per min., instead of more. When the apparent heat absorbed by the circulating water (taken in pounds per hour and temperature rise) was checked against the heat given up to the water by the steam condensed, an error was found which verified the foregoing. The corrections applied to the observed vacuum are, therefore, far from partial.

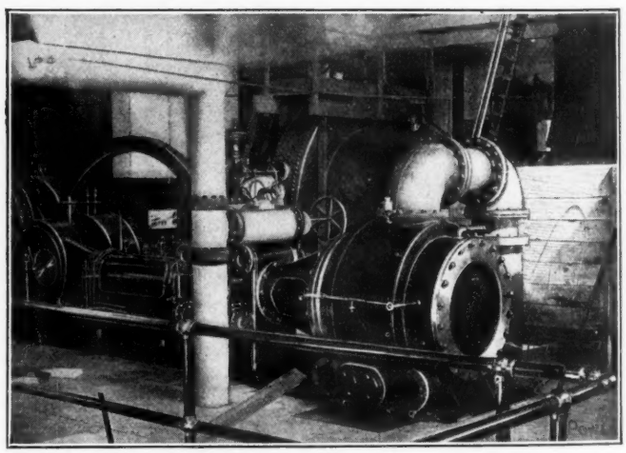


FIG. 4. ROTATIVE DRY-AIR PUMP

In Table 2, summarizing the tests, the following items should be noted: The remarkably high vacuums obtained with cooling water at 73½ deg. temperature; the small difference in temperature between the discharged condensing water and the steam, that is, from 4 to 10 deg.; the high hotwell temperature, ranging only from 2 to 6 deg. below steam temperature, and the high coefficient of heat transmission: 550 B.t.u. per sq.ft. per hour per

degree difference in temperature with a load of 210,000 lb. of steam per hour.

The greatest load on the condenser was 210,000 lb. of steam per hour, but it is evident from the test that a load of 250,000 lb. of steam per hour would have been

TABLE NO. 2. TEST OF WHEELER DRY TUBE SURFACE CONDENSER. ILLINOIS STEEL COMPANY — JULY, 1910.

Average vacuum referred to 30" barometer	Temp. corresponding to vac.	Lb. steam per hr.	Hot-well temp.	Temp. inlet circ. water	Temp. outlet circ. water	Rise in temp. of circ. water	Coeff. of heat trans. B.t.u. per h. per sq.ft. per deg.diff. (Based on 980 B.t.u. per lb. of condensate)
28.246	96.9	196,000	90.6	73.40	86.1	12.70	470
28.250	96.8	187,000	90.5	73.40	85.5	12.1	440
28.310	95.7	167,000	89.4	73.40	84.4	11.0	403
28.394	94.05	169,000	88.7	73.40	84.6	11.2	462
28.374	94.50	171,000	88.8	73.40	85.2	11.8	466
28.331	95.30	190,000	89.4	73.40	85.8	12.4	500
28.304	95.80	197,000	89.8	73.40	86.2	12.8	509
28.290	96.08	204,000	90.1	73.40	86.6	13.2	528
28.286	96.15	210,000	90.3	73.40	86.9	13.5	550
28.311	95.70	190,000	89.3	73.40	85.5	12.1	480
28.380	94.32	187,000	89.2	73.40	85.4	12.0	519
28.361	94.75	190,000	89.4	73.40	85.8	12.4	522
28.443	93.05	175,000	89.1	73.40	84.2	10.8	504
28.473	92.50	171,000	87.7	73.50	84.6	11.1	532
28.501	91.85	167,000	86.2	73.70	84.1	10.4	535
28.510	91.70	163,000	87.5	73.70	84.0	10.3	526
28.551	90.75	157,000	87.8	73.70	83.0	9.3	521
28.590	89.80	148,000	86.2	73.70	82.8	9.1	527
28.595	89.70	130,000	87.7	73.60	81.5	7.9	435
28.678	87.80	121,000	85.2	73.60	81.5	7.9	487
28.748	86.05	93,000	84.8	73.50	79.5	6.0	395
28.748	86.05	98,000	83.7	73.50	80.5	7.0	447
28.786	84.90	98,000	83.7	73.60	80.1	6.5	500
28.803	84.70	91,000	83.3	73.60	80.0	6.4	478

readily carried with 28-in. vacuum and 70-deg. water. This is equivalent to 10 lb. of steam per sq.ft. of surface, and under these conditions, the coefficient of heat transmission would have been about 600.

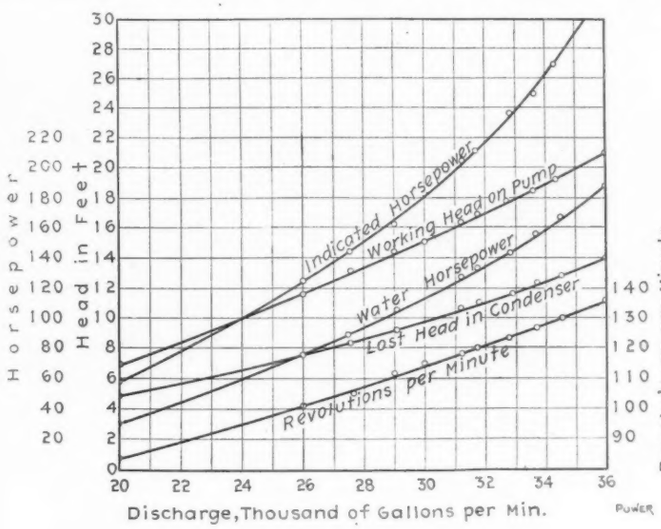


FIG. 5. SHOWING LOSS OF HEAD

The coefficients of heat transmission are calculated on the basis of 980 B.t.u. per pound of condensate, and by the logarithmic formula for heat transmission, which follows:

U = Coefficient of transmission in B.t.u. per hour per sq.ft. per degree difference in temperature;

W = Pounds of condensate per hour;

980 = B.t.u. per lb. of exhaust steam;

S = Square feet of condenser surface;

D = Mean temperature difference calculated by the following formula:

$$U = \frac{W \times 980}{S \times D}$$

Where

$$D = \frac{t_2 - t_1}{log.e \dfrac{t_s - t_1}{t_s - t_2}}$$

in which

t_s = Steam temperature corresponding to the vacuum;

t_1 = Inlet condensing water temperature;

t_2 = Outlet condensing water temperature;

$log.e$ = Natural logarithm.

It will be of interest to compare the heat transmission obtained in this condenser under actual operating conditions, with the heat transmission obtained in tests on experimental apparatus. The average water velocity in these tests was about 4.7 ft. per sec. Referring to the *Transactions* of the A. S. M. E., December, 1910, and the experiments by George A. Orrok on a condenser containing one tube, we find that for a circulating-water velocity of 4.7 ft. per sec., the coefficient of heat transmission as given by him is about 660 B.t.u. per sq.ft. per degree difference in temperature per hour. This coefficient is but slightly higher than the coefficient 550 obtained above.

TEST ON AIR PUMP AND CIRCULATING PUMP

In the test on the dry-vacuum pump, Table 1, attention is drawn to the average steam and air cylinder indicated horsepower and to the remarkably high mechanical efficiency.

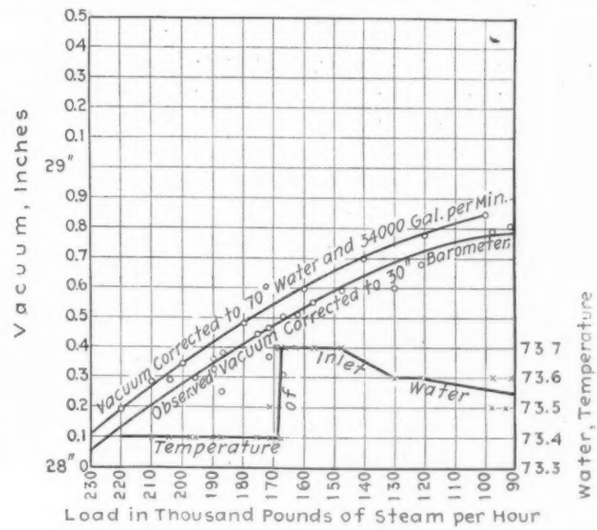


FIG. 6. VACUUM OBTAINED WITH WATER AVERAGING 73.6 DEG. F.

A test on the centrifugal circulating pump is covered by the charts of Figs. 5 and 7. By plotting the curve of head lost in the condenser against the discharge in gallons per minute on logarithmic paper, it is found that the head lost varies as the 1¾ power of the discharge; but since the velocity varies directly as the quantity of the water, we conclude that the head lost varies according to the 1.75 power of the velocity. This exponent is but slightly lower than the exponent 1.87 given in the American Civil Engineers' handbook, or than that

of 2, the exponent generally conceded as an average value. The exponent of 2 was checked several years ago by Prof. Josse and was found consistent with results obtained with a 300-kw. condenser.

As the water horsepower depends upon the quantity

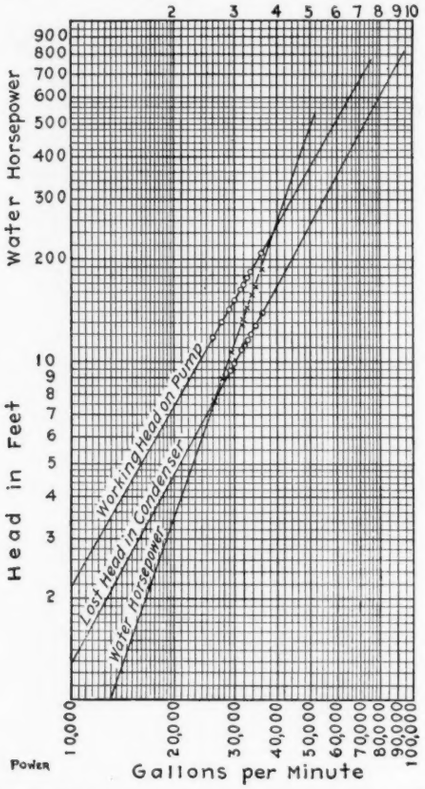


FIG. 7. CURVES OF CENTRIFUGAL-PUMP TEST

times the head, the horsepower should vary as the cube of the velocity. It is found by plotting the water horsepower curve of Fig. 6 on logarithmic paper (Fig. 7) that the exponent is 2.9; that is, the horsepower varies as the 2.9 power of the gallon per minute, or of the velocity through the condenser tubes.

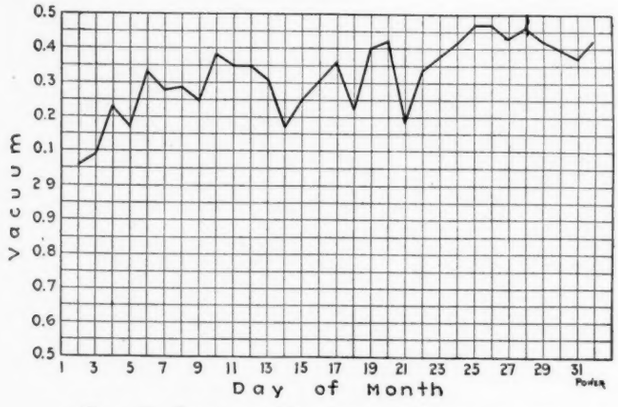


FIG. 8. LOG OF MONTHLY PERFORMANCE

A log of a month's performance of this condensing outfit is shown graphically by the chart of Fig. 8. This log covers performance in the month of January with the inlet cooling water down to about 34 deg. F. The performance is markedly consistent. The maintenance of vacuums of 29.3 in. and 29.4 in., day in and day out is what produces substantial savings in coal consumption.

Vitribestos Stack Lining

The smoke-stack of the First National Bank Bldg., Pittsburgh, Penn., which is 400 ft. high, is lined throughout with vitrified asbestos or "Vitribestos" to a thickness of 2 in. and presents a solid interior surface. It serves the double purpose of protecting the metal stack from sulphurous and other destructive acids and insulating it from great heat. These slabs of vitribestos, 3 ft. by 6 ft. and 2 in. thick, weigh, with all attachments, only 30 lb. per cu.ft. in place, or about 5 lb. per sq.ft. of stack area.

This building was designed to be 26 stories high, but only five stories were built in the original undertaking completed four years ago, and the original stack was lined as stated. When the remaining stories were completed, the stack was extended in 18-ft. sections, riveting all joints except at the ends of the sections, which were flanged joints. The 18-ft. sections were lined with Vitribestos before they were carried to the building, and were hoisted and set in place without interferring with the operation of the stack. When the last section was in place, the stack was lined, except at the open joints, every 18 ft.

With the arrival of mild spring weather, these joints were sealed up and inspection showed that the material which had been in service for four years was in as good condition as when first applied.

This vitribestos is manufactured by the H. W. Johns-Manville Co., of New York City.

BOOKS RECEIVED

THE "MECHANICAL WORLD" ELECTRICAL POCKET BOOK FOR 1913. Emmott & Co., Manchester, England. Cloth; 311 pages, 4x6 in.; illustrated; tables. Price, sixpence.

THE INDICATOR HANDBOOK. Part II. By Chas. N. Pickworth. Emmott & Co., Manchester, England. Cloth; 139 pages, 4¾x7¼ in.; illustrated; tables; indexed. Price, three shillings.

STEAM BOILERS. By E. M. Shealy. McGraw-Hill Book Co., New York. Cloth; 356 pages, 6x9½ in.; 185 illustrations; tables; indexed. Price, $2.50.

THE ELEMENTS OF HEATING AND VENTILATION. By Arthur M. Greene, Jr. John Wiley & Sons, New York. Cloth; 324 pages, 223 illustrations; tables; indexed. Price, $2.50.

TEXTBOOKS OF SCIENCE—The Elements of Machine Design. Part II. New and revised edition. By W. Cawthorne Unwin and A. L. Mellanby. Longmans, Green & Co., New York. Cloth; 426 pages, 5½x9 in.; 311 illustrations; indexed. Price, $2.50.

PRACTICAL MATHEMATICS. Part III. By Claude Irwin Palmer. McGraw-Hill Book Co., New York. Cloth; 178 pages, 4½x7 in.; 31 illustrations; indexed. Price, 75c.

WINDING ENGINES AND WINDING APPLIANCES. By George McCulloch and T. Campbell Futers. Longmans, Green & Co., New York. Cloth; 452 pages, 5½x9-in.; 175 illustrations; tables; indexed. Price $6.

WATER, ITS PURIFICATION AND USE IN THE INDUSTRIES. By W. W. Christie. D. Van Nostrand Co., New York. Cloth; 219 pages, 5¼x8-in.; 79 illustrations; tables; indexed. Price $2.

ESSENTIALS OF ELECTRICITY. By W. H. Timble. Wiley & Sons, New York, and Chapman & Hall, Ltd., London. Cloth; 271 pages; 7⅛x4⅝ in.; illustrated; tables; indexed. Price, $1.25.

ANNUAL REPORT OF THE SMITHSONIAN INSTITUTION, WASHINGTON. Government Printing Office, Washington, D. C. Cloth; 688 pages; 9¼x6 in.; illustrated; tables; indexed.

AMERICAN WATER WORKS ASSOCIATION PROCEEDINGS, 1912. By the Association. Waverley Press. Cloth; 465 pages, 9¼x6½ in.; illustrated; tables; indexed.

ELEMENTS OF HEAT-POWER ENGINEERING. By Hirshfeld and Barnard. Wiley & Sons. Cloth; 811 pages, 9⅜x6¼ in.; illustrated; tables; indexed. Price, $5.

HANDBOOK OF ENGLISH FOR ENGINEERS. By Wilbur Owen Sypherd. Scott, Forsman & Co., Chicago and New York. Leather; 314 pages, 7x4¾ in.; indexed. Price, $1.50.

NEW PUBLICATIONS

STEAM ECONOMY IN THE SUGAR FACTORY. By Karl Abraham. Translated from the German edition by E. J. Bayle. John Wiley & Sons, New York City. Cloth; 104 pages, 5¼x7¾ in.; 15 tables. Price, $1.50.

The author, who is general engineer of the American Beet Sugar Co., translated the present book so that the American manufacturer might have the advantage of European practice concerning the proper distribution and management of steam to effect economies in the production of sugar from beets. Under a division heading of the Steam Consumption of the Individual Stations the author discusses diffusion, heaters, saturation of thin juice, steam consumption for heating the juice and for the evaporation proper, boiling, heating of the thick juice and syrups, treatment of the green syrups and the remelted products, turbinating and washing, steam losses due to radiation, and mechanical work. The second division of the book deals with the distribution of steam in the factory, covering methods of reduction of steam consumption, adaptability of different systems for utilizing juice vapors, distribution of temperature in the evaporating system, diagrams for the distribution of steam, influence of methods of operation on total steam consumption, calculation of heating surfaces. To help toward a better understanding of the text matter numerous problems are presented and the solutions given, and data and tables have been converted into English standards. The book should prove of value to engineers and designers of sugar plants.

MACHINE DESIGN—HOISTS, DERRICKS, CRANES. By H. D. Hess. J. B. Lippincott Co., Philadelphia. Cloth; 368 pages, 6x9½ in.; 318 illustrations and 18 plates; tables. Price, $5 net.

This book, although primarily intended to aid technical students in the work of machine design, should prove useful to anyone directly engaged in the design or manufacture of this class of machinery. The book is divided into twelve parts: Introduction, Frames and Girders, Brakes and Clutches, Winches and Hoists, Pillar Cranes, Jib Cranes, Under-braced Jib Crane, Inverted Post Crane, Wall Crane, Overhead Electric Traveling Cranes, Hoisting Engines, Locomotive Cranes.

The introduction includes a discussion of properties of materials and fiber stresses, the design of machine elements such as springs, belts, pulleys, gearing, shafting, bearings, chains, drums, crane hooks, etc.; elements of graphic statics, rolling and sliding friction, crane blocks, wire ropes, structural members and rivets.

In the other chapters suitable data are selected and the problems worked out, the illustrations being drawn from present-day American practice. The part devoted to electric traveling cranes is especially good. The treatment is simple and by applying elementary principles in determining the stresses in frames and machinery it is shown that the design may be carried out on a theoretical rather than on an empirical basis. The method of procedure in applying the principles of design to practice is clearly outlined, so that the book should be of great assistance in directing the student and serve as a valuable reference to the designer.

THE GAS ENGINE. By Hans Holzwarth, Engineer. Translated by A. P. Chalkley, B. Sc., Am. Inst. C. E., A. I. E. E.; 6x9 in.; 40 pages, illustrated, cloth. Charles Griffin & Co., Ltd., London. J. B. Lippincott Co., Philadelphia, 1912. Price, $2.50.

In our issue of February 6 last we described at considerable length a 1000-horsepower gas turbine, designed by Hans Holzwarth and built by Brown, Boveri & Co., at Mannheim. In this article we referred to a book recently published in German by Mr. Holzwarth which furnished largely the information upon which our article was based. The translation under review makes this work available in its entirety to English-speaking engineers.

The book is divided into four parts, the first of which is a highly mathematical consideration of the theory of the subject. Part II, entitled "Construction of the Gas Turbine," deals entirely with the details of the construction of the 1000-horsepower machine of the author's design above referred to.

Under the caption "General Comparison," Part III compares the gas turbine and the reciprocating gas engine from a thermodynamical standpoint, leading up to the conclusion that the gas turbine cycle which the author employs is not similar to the Lenoir cycle, but is much more comparable with the cycle of the free piston engine which the Humphrey gas pump resembles. A comparison with the reciprocating gas engine is then taken up from the point of view of construction and operation. Under the caption "Gas Turbine Plant" a proposed installation of three 1000-horsepower gas-turbine-driven units is described.

Part IV gives results of tests of the original gas turbine built to the design of the author by Körting Brothers of Hanover and the 1000-horsepower machine built by Brown, Boveri & Co. While these tests were disappointing, both as to capacity and efficiency, the reasons for their limitation are believed by the author to be sufficiently evident to indicate the direction in which future effort should be directed and so apparent that the realization of the author's expectation is already within sight.

STEAM BOILERS AND BOILER ACCESSORIES. By W. Inchley, lecturer on engineering at the University College, Nottingham, England. Published by Longmans, Green & Co., New York and London, 1912. Size, 5x7½ in.; illustrated; cloth. Price, $2.40.

In this book an attempt is made to treat the subject of steam boilers and boiler accessories in a manner suitable for the requirements of steam users and engineering students.

After the function of the steam power plant is explained, the various types of boilers are briefly considered. To bring out the advantages of the various types of boilers, Table 1 is given, which includes, for every type illustrated, numerical values for the pounds of water evaporated per hour from and at 212 deg. F., heating surface, grate area, ratio of the heating surface to the grate area, pounds of water per hour per square foot of the grate area and of the heating surface, and dimensions of the setting. Materials for boiler construction are then taken up, and, in connection with the subject of the brittleness of steel, special attention is called to the effect of phosphorus and nitrogen on the ultimate strength of the boiler plate. Table 2 gives 30 cases of failure in steel plates and the per cent. of carbon, nitrogen and phosphorus in each case. At the end of the chapter formulas for calculating the strength of the boiler shell are derived.

The fuels for the generation of steam are treated rather too briefly, but the subject of combustion is taken up at considerable length and in a very clear manner. Methods for calculating the air required for combustion from flue-gas analysis and the subject of coal calorimetry are considered in detail. At the end of the chapter several practical numerical examples are given.

Boiler furnaces are treated at considerable length. The various losses which take place in a boiler furnace are considered and the methods of decreasing them outlined. The conditions affecting the design of furnaces for various fuels and for different kinds of boilers are also discussed. Various types of English stokers are illustrated; these differ much from those used in American power plants. The subject of combustion of liquid and gaseous fuels is also treated at some length.

The fundamental theory of the chimney draft is explained and formulas are derived for calculating the height of a chimney to produce a given draft. In connection with the subject of mechanical draft steam jets are compared with fans and the type of boiler best suited for mechanical draft is discussed. More might have been given on mechanical draft and additional illustrations would have been desirable.

The theory of steam generation is simply and clearly explained and considerable space is given to discussing the effect of the velocity of gases upon conduction and on ultimate boiler performance. This section is well written and uptodate, except that the old values for the properties of saturated steam were retained. This might tend to confuse the reader, as the latent heat of steam at 212 deg. F. is given as 966.6 B.t.u. in the text and as 970 in the steam tables at the end of the book.

Much space is given to the details of the Lancashire and other flue boilers, and but brief mention is made of the cylindrical multitubular type. The locomotive and vertical boilers are treated briefly and do not include uptodate designs in American practice.

Water-tube boilers are covered very completely, but also represent mainly English practice.

In connection with boiler accessories, feed pumps, injectors, water and oil filters, steam and oil separators, feed-water heaters and economizers, feed-water regulators, reducing valves and superheaters are described in detail. Many good illustrations are included. The treatise on superheaters is especially complete.

In connection with steam piping the subjects of water-hammer, drainage, protection against expansion and vibration, corrosion and steam-pipe covering are considered.

The selection and treatment of boiler feed water is dealt with at considerable length and includes methods of testing water for corrosiveness and means of estimating and remedying hardness of water.

Directions for the practical running of boilers are given,

including an explanation of the use of CO_2 recorders and directions regarding the purchase of fuel.

Under steam-boiler testing the code of the Institution of Civil Engineers of England is given. This differs from the code of the American Society of Mechanical Engineers mainly in the details regarding the recording of results. The instructions regarding the duration of the test, the methods of sampling the fuel and ashes and the analysis of coals are presented very clearly. Also the directions for working up boiler tests are given in great detail.

In general the book is clear, contains much new matter, and, if edited so as to include American practice, would prove a very desirable textbook for colleges. In its present form it is a good reference book and should prove of considerable value to engineers.

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Lectures on Engineering Chemistry

The first of a series of three lectures by Dr. Frederick Dannerth, consulting chemist, on "Some Phases of Engineering Chemistry," was delivered before the Institute of Operating Engineers, at its quarters in the Engineering Societies Building, 29 West 39th St., New York City, on Saturday evening, Feb. 15. This lecture related chiefly to coal; its combustion and purchase by specification. The two forthcoming lectures, the dates of which have not yet been announced, will treat the subjects of boiler feed water and oils and lubrication.

SOCIETY NOTES

The Iron and Steel Institute, London, will hold its annual meeting at the Institution of Mechanical Engineers, Westminister, on May 1 and 2. At this meeting the Bessemer gold medal will be awarded to Adolphe Greiner, general director of the Société Cockerill, Seraing, vice-president of the institute. The annual dinner will be held at the Hotel Cecil, on May 1. The autumn meeting, at Brussels, will be announced later.

A reception and ball by the American Institute of Electrical Engineers, at the Hotel Astor, New York, will be the closing feature of its midwinter convention, beginning at 9 p.m., Feb. 28. As it takes the place of the annual banquet usually held in February, it will be the social event of the season, giving every opportunity for the greeting of old friends and making new acquaintances. Tickets to the reception, at $2.50 each, may be obtained from Ralph W. Pope, honorary secretary of the institute, or at the information bureau of the convention at the Engineering Societies Building, 33 West 39th St., New York. Checks made payable to R. W. Pope should accompany all requests.

OBITUARY

VICTOR HUGO

Victor Hugo, about 14 years manager and chief inspector of the St. Louis department of the Hartford Steam Boiler Inspection & Insurance Co., died very suddenly on Jan. 31 from pneumonia. He was a graduate of the University of Minnesota, a member of the American Society of Mechanical Engineers, and a man whom it was good to meet and to know. His loss will be seriously felt by all with whom he was connected.

JOHN FRITZ

On Feb. 13 John Fritz, famous mechanical and mining engineer died at his late home Bethlehem, Penn., at the age of 90 years.

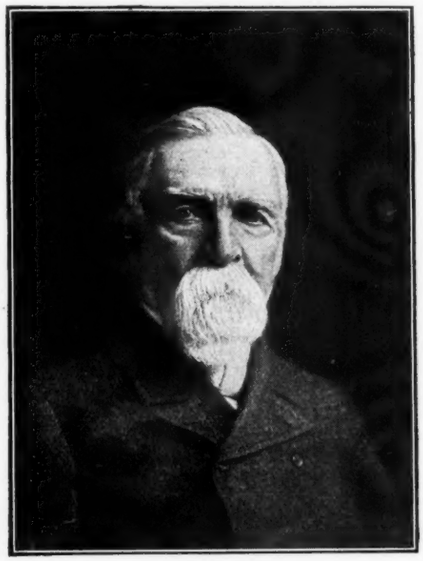
Mr. Fritz, known among engineers as "Uncle John," was born Aug. 21, 1822 in Londonderry Township, Chester County, Penn.

His early boyhood was spent on his father's farm, where he received what little education the local school offered. At the age of sixteen he became an apprentice in a machine shop, and later entered the services of the Moore & Hoover Iron Works. His mechanical proclivities enabled him to perceive the great possibilities of the iron industry. While employed there he began to specialize in machine design. From the Moore & Hoover company he went with the Morristown Iron Works, and later to the Cambria Iron Works of Johnstown, Penn.

While with this company his achievements in iron rail production were such as to attract wide attention. He revolutionized the making of iron rails, effecting a better quality and a production four-fold greater than any prevailing methods, thus enabling American manufacturers to compete with those of England and Wales. It was about this time that he also introduced improvements in blast furnace practice.

In July, 1860 he began service with the Bethlehem Iron Co. as general superintendent and chief engineer. While there he greatly assisted in perfecting the bessemer process of making steel, and also in remodeling and improving the Siemens-Martin open-hearth process. Turning his attention to structural and plate mill practice he designed mills and rolls which turned out beams, channels, etc., superior to those on the market. When the problem of making armor plate arose, he originated the solid steel plate, the superiority of which over all other forms remains unquestioned.

The success and growth of the great Bethlehem Iron & Steel plants are due largely to the efforts of Mr. Fritz, whose



JOHN FRITZ

engineering ability and accurate judgment were always productive of commendable results.

He was a member of all the principal engineering societies of this country and abroad, and was honored by many of the scientific societies of the world. The banquet given at the Waldorf-Astoria on Oct. 31, 1902 on the occasion of his eightieth anniversary and the foundation of the John Fritz medal to perpetuate his memory, was one of the most felicitous occasions in the history of American engineering.

The engineering profession has lost one of its great members, and engineers and scientific men throughout the world will mourn the loss of one whose name will forever be associated with the steel industry.

PERSONALS

F. R. Wadleigh has opened offices at 1013 Bank of Commerce Building, Norfolk, Va., as consulting engineer, on all matters connected with the preparation, purchase and use of coal. Mr. Wadleigh will give especial attention to sampling and analyses, and use of coal as fuel.

E. A. Scott, managing editor of the "Metal Worker," has been appointed secretary of the American Society of Heating and Ventilating Engineers. W. W. Macon, who has held this position for a number of years, is now a member of the board of managers.

F. William Bösch has resigned his position in the steam turbine drafting department of the General Electric Co., Lynn, Mass. After a visit to Germany, where in connection with Prof. Rössler, Mittercide, he will make a study of high-speed engines, especially investigating the Stumpf uniflow steam engine, he will, in August, assume the position of designing and consulting engineer with the Murray Iron Works Co., Burlington, Iowa.